BELT CONVEYORS

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BY

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PREFACE TO THIRD EDITION

This has been a standard work on the subject ever since its publication in 1922; because the second edition of 1926 has been out of print for several years past, it was decided to issue a third edition. The preface to the first edition said, "This is intended to be a practical book. It is not a mere restatement of what already appears in trade advertisements, nor does it contain descriptions of installations of conveying and elevating machinery. It aims rather to explain principles and the reasons for doing things."

It has been the purpose of the authors to follow the same principles in this present third edition, telling what is new in present-day practice, what is better and why, and what has become obsolete and why.

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SECTION I. BELT CONVEYORS

CHAPTER 1

GENERAL DESCRIPTION OF COMPONENT PARTS

A belt conveyor consists of a moving endless belt which supports material and which by its motion carries the material from one place to another. The belt is driven by a pulley and is supported on both runs, going and coming, by rollers or by a runway. The material may be put on the belt by hand, shovel, chute, or some other means, and it is removed from the belt by discharging it over the end pulley or by deflecting it at some point along the run of the conveyor.

The elements of a belt conveyor are, therefore:

1. A belt to carry the material and transmit the pull.
2. Means to support the belt, usually rollers or pulleys.
3. Means to drive the belt, usually a pulley or a pair of pulleys.
4. (*a*) Accessories for maintaining belt tension, such as take-ups.
   (*b*) Accessories for loading the belt, such as a chute.
   (*c*) Accessories for discharging the material, such as a chute or a tripper, or a scraper.
   (*d*) Accessories for cleaning and protecting the belt, such as housings, decks, covers, and cleaning brushes.

The belt is a flexible jointless structure which runs quietly at any speed; it is not ordinarily harmed by the actual conveying of the material it carries. Since the material does not come into contact with the moving surfaces of pulleys and shafts in which there are friction losses, these losses are relatively small and the power required for the transfer of material is generally less than in other forms of conveyors. The belt with its rollers weighs less per foot of run than other types of conveyors doing the same or similar work, and hence frames, bridges, and other supporting structures are relatively lighter and cheaper.

Belt conveyors are suited to the carrying of all sorts of material, wet or dry, from the lightest to the heaviest, and in any quantity.

They have been known and used for over a hundred years, but the most rapid development in their design and use has occurred since 1893.

The Belt. The belt must have a certain flexibility in order to wrap around the pulleys, width enough to carry the required quantity of material, and strength enough to bear the weight of the load and transmit the pull in the conveyor. These conditions can be met by bands of metal, leather, or woven fabric. For metal belts see page 78. Leather belts are expensive and do not resist wet and abrasion well enough in conveyors and elevators to justify their greater cost.

Belts of hemp fiber are used to some extent in Europe, but in this country most of our conveyor and elevator belts are made of cotton fiber. They are of several forms:

1. Rubber belts are made of layers or plies of cotton duck cemented together by an elastic rubber compound. In "friction surface" belts the outside of the belt is covered by the thin layer of compound adhering to the outer plies; in rubber-covered belts an extra layer of rubber compound is attached to the outer plies beyond the thin coating of "friction rubber." No attempt is made to waterproof the individual cotton fibers, the layers of rubber being depended upon to keep moisture out of the belt. See page 34.

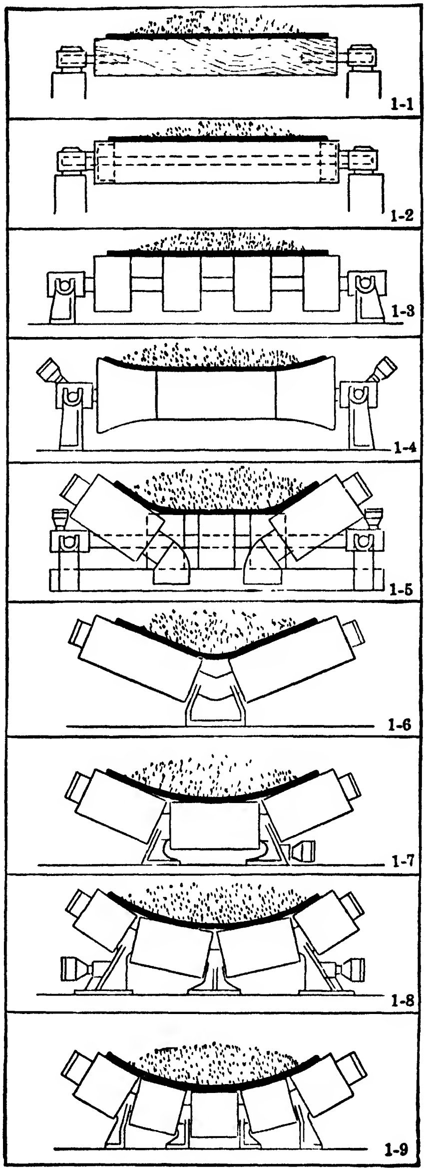
2. Stitched canvas belts are made of layers or plies of cotton duck folded together to give the required width and thickness, and then sewed through and through with strong cotton twine. To waterproof the fibers and to reduce internal wear, the made-up belt is impregnated with a mixture of oil and gum. See page 60.

3. Balata belts are made of duck with the fibers of the cotton waterproofed by impregnation with a liquid solution of balata, a tree gum similar in some ways to rubber. The impregnated duck is folded and rolled under pressure to make a belt of the required width and thickness, the balata gum acting as a cement to hold the plies together. See page 68.

4. Solid-woven belts consist of a number of layers of warp (lengthwise) threads and weft or filler (crosswise) threads woven and interbound together in a loom to make a structure of fabric of the necessary width and thickness. Most of them are waterproofed like stitched canvas belts, but some are impregnated with a rubber solution and then covered with a rubber sheathing. See page 66.

Supports for the Belt. Supporting idlers consist of rollers or pulleys, of wood, cast iron, or steel in various forms and combinations. For light work, especially in handling packages, the roller may be a cylinder of hard wood (Fig. 1–1), or a piece of thin steel

tubing with inserted heads of wood or metal (Fig. 1–2). For heavier work, the roller may consist of several cast-iron pulleys mounted on a through shaft (Fig. 1–3), or the outer pulleys of the combination may be enlarged at their outside ends (flared idlers) in order to lift the edges of the belt slightly and prevent spilling (Fig. 1–4). When it is desired to increase the carrying capacity of a belt it may be bent into trough form by turning up the edges of the belt by troughing or "concentrator" pulleys (Fig. 1–5), or by supporting the whole width of the belt on idlers in which two, three, four, or five pulleys are set at various angles from the horizontal to trough the cross section of the belt (Figs. 1–6, 1–7, 1–8, 1–9).



FIGS. 1–1 to 1–9. Supporting Idlers for Loaded Run of Conveyor Belts.

Driving the Belt. The drive for a belt conveyor consists of one or two pulleys around which the belt wraps, suitably mounted on shafts and bearings and driven from a source of power through belts, chains, gears, or other means of power transmission. The simplest drive is that in which the belt wraps halfway round the end pulley; this is usually at the head end or delivery end of the conveyor toward which the material moves, but it may be at the foot or loading end of the conveyor (Fig. 1–10). If 180° of belt wrap is not enough to drive the conveyor, it may be necessary to get a greater wrap on the driving pulley by means of a snub or reverse-bend pulley (Fig.

1–11). For still greater driving contact, the belt may be led around two pulleys, both of which are drivers. Figs. 1–12 and 1–14 show two arrangements of these tandem-drive pulleys. To accomplish the same result, a pressure belt (Fig. 1–13) may be used to increase the grip between a conveyor belt and a single drive pulley

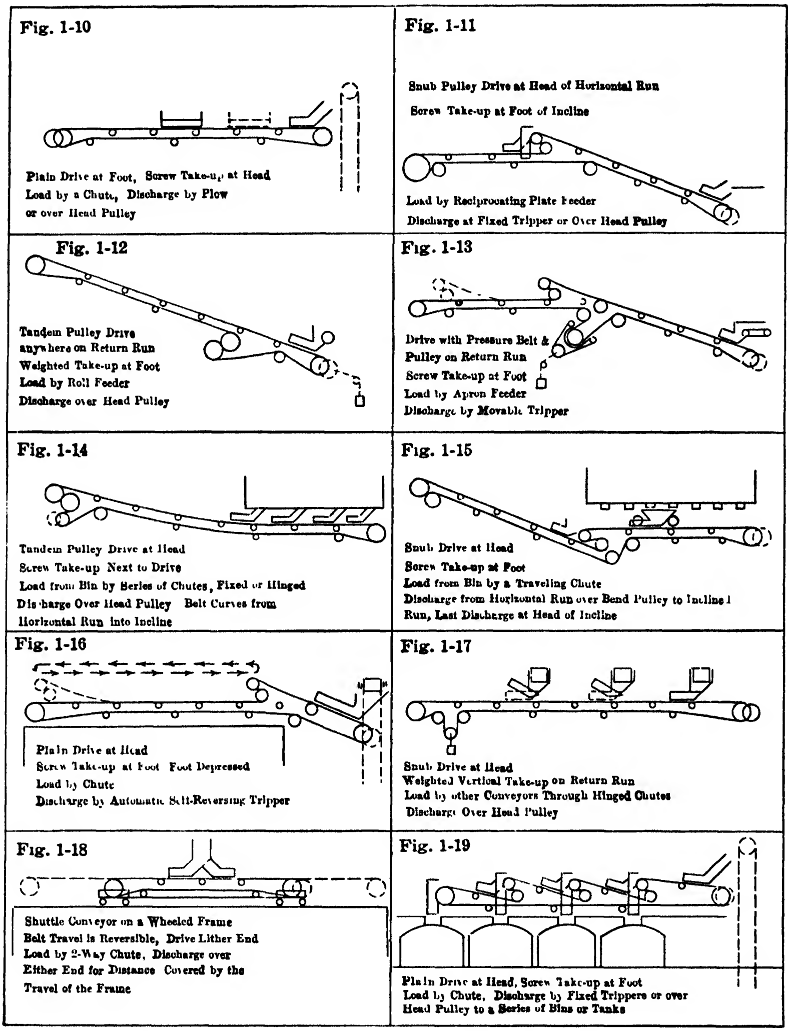
Belt-conveyor Accessories. Conveyor belts in service stretch more or less, and it is necessary to have means to "take up" or remove the slack as it is formed. At some point in the conveyor, a pulley is mounted on a shaft running in bearings which are adjustable in position either by a screw or by a weight. Fig. 1–10 shows a conveyor with a screw take-up at the head end; in this case the tension required to pull the loaded side of the belt is transmitted through the take-up; the usual position for the take-up is at the foot end as in Fig. 1–11, where the belt is under less tension. In Fig. 1–14 the take-up is at the point of *least* tension, that is, where the belt runs off from the driver. Fig. 1–12 shows a take-up shaft with a pull-back weight; in Fig. 1–17 a "gravity" take-up on the return belt takes care of the slack and maintains driving contact on the end pulley.

Loading. Loading chutes may be used in connection with a gate controlled by hand, as in drawing grain from bins, or to deliver to the belt a supply of material at a uniform rate as in Figs. 1–10 and 1–14. If the material is hard to control by a gate, or if the supply is intermittent or irregular, feeders of various kinds (Figs. 1–11, 1–12, 1–13) are used in connection with loading chutes.

Discharging. The simplest discharge is over the end pulley (Figs. 1–12, 1–17). Sometimes a chute may be required there; often it is not. If the discharge is to be at some point short of the end of the conveyor, the material may be deflected sideways from the conveyor by a plow or scraper set diagonally across the belt (Fig. 1–10); more frequently this is done by inverting the belt or running it in S form through a tripper (Figs. 1–11, 1–13, 1–16, 1–19). In the tripper or "throw-off," as it is called in England, the material leaves the belt as it reaches the top of the upper pulley and is caught in a chute which directs it to one side, or by means of a by-pass gate, back on the belt again if the material is to be carried past the tripper as in Figs. 1–11 and 1–19. Trippers may be fixed (Fig. 1–11), or movable (Fig. 1–13), or traveling and self-reversing (Fig. 1–16), and may be operated by hand or power.

Protecting the Belt. The belt is the most expensive part of a belt conveyor, often costing more than all the rest of the machinery and accessories combined. At the same time it is the most vulnerable part; it is subject to abrasion from impact of the material and to a

number of injuries from negligence or accident. Injury at the loading point can be avoided by proper design of the chute; other mishaps can be prevented by care in operation and maintenance. To lessen the



FIGS. 1-10 to 1-19. Typical Belt Conveyors with Various Arrangements of Drive, Feed, Discharge, and Take-up.

risk of certain other injuries it is necessary to provide means to clean the belt from adhering particles, to cover it where it is exposed to the weather, and to prevent it from being cut by objects falling against it or upon it.

CHAPTER 2

DEVELOPMENT OF BELT CONVEYORS

History in the United States. The earliest reference to the use of belt conveyors in American practice is in Oliver Evans' "Miller's Guide" published in Philadelphia in 1795. This describes and illustrates a flat belt receiving material on its upper run and discharging over the end; Evans calls it "a broad endless strap of thin pliant leather or canvas revolving over two pulleys in a case or trough." In the flour and grist mills built by Evans and his successors some of these conveyors were probably used, but it was more common in those days to convey grain as well as the lighter mill products by means of screw conveyors.

Belts sliding in troughs (Fig. 2-1) were used before 1840 to convey materials which were not suitable for screw conveyors, such as clay,

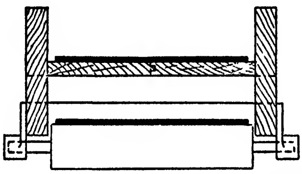


Fig. 2-1. Conveyor Belt Sliding in Wood Trough, 1830.

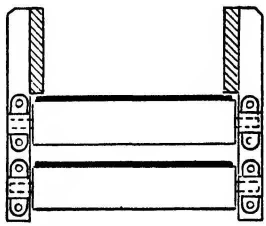


Fig. 2-2. Clay Conveyor Belt Supported on Rollers, 1870

shavings, sawmill refuse, and crushed stone. Sometimes the return run slid back; sometimes it was carried on rollers. In some cases the sides of the trough were made with hinged sections which swung inward to stand diagonally across the belt and thus discharge material along the run of the conveyor.

When such conveyors were used for hard and gritty substances, the belt did not last long. In clay conveyors, the clay stuck to the sides and bottom of the trough, hardened there, and wore out the belt rapidly. Some of these troubles were avoided in the design shown in Fig. 2-2, which represents a form of clay conveyor in extensive use before 1870. This construction substituted rolling friction in lubricated bearings for sliding friction on dirty rough surfaces; it saved power

and helped to preserve the belt, although the material could not be carried without some leakage under the skirt boards or trough sides, and the edges of the belt were still subject to injurious wear.

The great increase in the quantity of grain handled in this country after 1850 and the development of the grain "elevator" or storage system created a demand for belts of larger carrying capacity. Merrick & Sons, of the Southwark Foundry in Philadelphia, who built the Washington Avenue Grain Elevator there between 1859 and 1863, used wide composite belts consisting of two parallel leather belts to which at intervals were riveted bent iron bars or spreaders to support a trough of canvas that carried the grain. This construction was similar to that shown in Fig. 2–3; power was applied to the edge belts by iron pulleys separated by the width of the canvas trough, or by a wooden drum grooved to clear the sag of the canvas. Narrow pulleys for idlers supported the edge belts on the loaded run and on the empty run.

Grain conveyors of this kind were installed in other American "elevators" during the sixties. As used at Duluth early in the seventies (T. W. Hugo, *Transactions A.S.M.E.*, 1884), they consisted of rubber belts 7 inches wide (Fig. 2–3) supporting a canvas trough 2 feet wide, which sagged 4 or 5 inches in the middle. They ran at 650 feet per minute and carried 12,000 bushels of wheat per hour. A composite belt of this kind had only 12 or 14 inches of belt width to engage the face of the driving pulley, and hence the length of the conveyor was limited to what that width of pulley face would drive; but, on the other hand, the load to be pulled corresponded to that of a belt 24 inches or more in width. Aside from that disadvantage the edge belts would not stretch alike, the spreader bars would tear loose from their fastenings, and when the bars pulled loose, the belts came off the idlers and there was serious trouble.

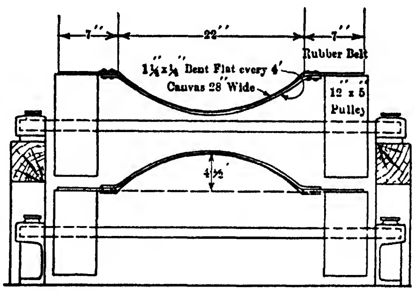


Fig. 2–3. Composite Belt for Grain Conveyor, 1870.

Invention of the Tripper. On February 10, 1863, United States patent 37,615 was issued to Oren C. Dodge of New York with the following claims:

> 1. Delivering the grain at any desired point along the line of a traveling belt by bending such belt substantially as specified, for the introduction of a hopper or chute.

2. A traveling belt for conveying grain, provided with vertical or nearly vertical edges, forming a trough.

Dodge's drawings show an arrangement of fixed trippers and a flanged belt, with the belt supported on cylindrical rollers.

Grain Conveyors at Liverpool. About 1865, P. G. B. Westmacott and G. F. Lyster, engineers for the Birkenhead and Waterloo Docks at Liverpool, England, made experiments with a 12-inch belt which showed that belt conveyors carried more grain with less power than screw conveyors. In 1866 they installed a system consisting of a skip hoist for elevating the grain with belts for distributing it. The belts were 18-inch 2-ply rubber, run at 450 to 500 feet per minute and supported every 6 feet on straight wooden rolls. Discharge was effected by running the belt through a traveling tripper pushed by hand. Tuck-up or concentrator rolls mounted on portable frames were used to keep the grain back from the edge of the belt at the loading point and where the belt began to lift into the tripper. How much Westmacott and Lyster knew about American practice at that time we cannot say. Westmacott's account of their work (*Proceedings Inst. of Mech. Engrs.*, 1869) makes no reference to the prior use of belts for grain in the United States or to the use of trippers there. The traveling tripper was covered by Westmacott's British patent 3061 of 1866, and U. S. patent 66,759, in 1867; it was new and a great improvement over the fixed trippers shown in the Dodge patent.

Development of Grain Conveying. The improvements made at Liverpool were taken up by American engineers, at first by W. B. Reaney in alterations at the Washington Avenue Elevator in 1873 and in the design of a complete new Elevator (Canton 1) for the Northern Central Railroad at Baltimore in 1876. In this elevator, designed by Reaney and built by John T. Moulton & Son of Chicago, the conveyor belts were 30 inches wide 4-ply rubber and ran at 550 feet per minute over straight wood rolls placed every 5 feet on the carry and 10 feet on the return. Wooden concentrator pulleys similar to Fig. 2–4 were used at the loading points only. This work by Reaney introduced wide rubber belts into the business of handling grain. For a reference to a rubber belt which he installed at Philadelphia in 1873, see page 52.

The Canton 1 Elevator had the first traveling trippers in which the moving power was taken from the conveyor belt.

In other grain conveyors built in this country up to 1885, the belts were made to give greater capacity by troughing them over spool-shaped idlers (Fig. 2–4). As long as the difference between the middle and end diameters was only an inch or two, the pulley side of the belt

did not suffer appreciably from the fact that the edge of the idler traveled more than the middle in feet per minute and hence must rub the belt. When the trough was made deeper by making the sag of the belt 2 or $2\frac{1}{2}$ inches (see T. W. Hugo's paper), the wear on the belt was great enough to be noticed; but the main objection to the deep spool idler as used on some grain conveyors was that a belt lightly loaded or empty was likely to run crooked. This trouble always exists where all or a great part of the weight of belt and load is carried on revolving conical surfaces or on separate cylindrical surfaces which are set to trough the belt (see page 104). It led first to the use of guide pulleys to bear against the edge of the belt, always a dangerous and destructive practice; then to the "dishpan" idler, which came into use in the late seventies (Fig. 2–5). By having the end rolls or "dishpans" loose on the shaft and free to turn independently of the horizontal pulley, there was less chafing of the belt than with deep spool idlers as long as the belt had a slight contact with the end rolls (Fig. 2–5, left half). Under a heavy load, however, the belt assumes the shape shown in Fig. 2–5, right half, and the pulley side is rubbed so as to cause a noticeable loss of power in driving the conveyor. Besides that, there was difficulty in making the belt run straight in the narrower widths, and after 1890 this form of idler gradually became obsolete.

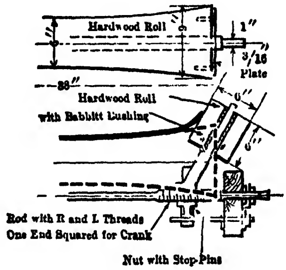


FIG. 2–4. Carrying Idler and Concentrator Roll 36-inch Grain Belt. (John T. Moulton & Son, 1880.)

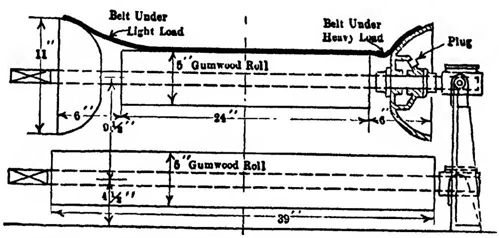


FIG. 2–5. "Dishpan" Idler and Return Idler for 36-inch Grain Belt, 1880.

After twenty years of experience with various kinds of belt idlers, designers of grain-elevator equipment in this country reached the conclusion that the right way to convey grain is by a belt that is flat (Fig. 2-6) or nearly so. A flat belt has a good contact with the horizontal idler pulleys and will run straight, empty or loaded, or

even loaded to one side of the center. Grain carefully fed to a flat belt will not tend to shift toward the edges and spill; but to prevent scatter at the loading points and to increase the carrying capacity

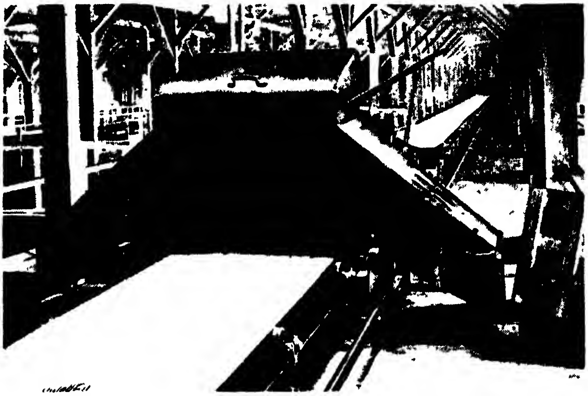


Fig. 2–6. Forty-inch Flat Grain Belt, 1897. (Link-Belt Company.)

moderately it is customary to use inclined concentrator pulleys at the loading points only, or at intervals along the length of the conveyor (Fig. 2–7). The first concentrators used in this country were inclined at 60° (Fig. 2–4), but as many cases of longitudinal crack-

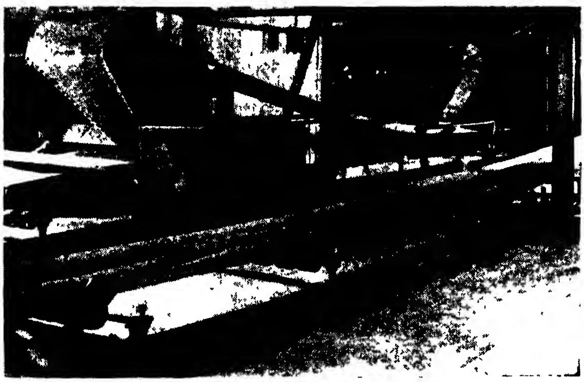


Fig. 2–7. Thirty-six-inch Grain Belts with Concentrators Every 15 Feet. (James Stewart & Co., 1908.)

ing of belts could be proved against the severe bend in the belt, the angle was reduced to 45°, and then to 35°, which is now most common. Concentrators with angles of 30° and 22° are also made, to reduce still further the flexure of the edges of the belt.

In modern practice the concentrators may be used on a separate stand (Fig. 2–8) or combined with the horizontal pulleys (Fig. 2–9), or a stand may be used to take the return idlers also.

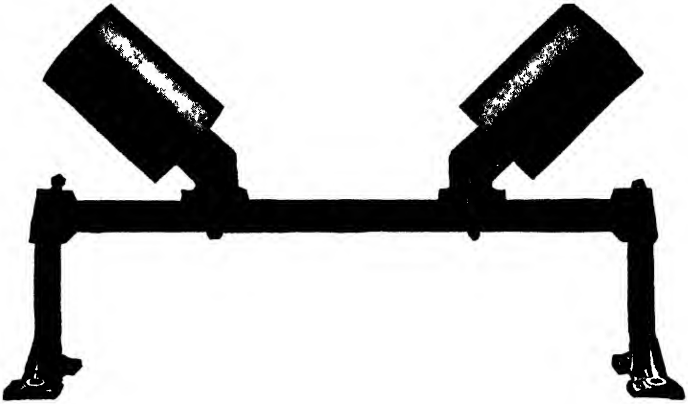


FIG. 2–8. Concentrator Pulleys Mounted on a Separate Stand.

Conveying Materials Heavier than Grain. With the development of conveying machinery between 1880 and 1890 belts were used for handling coal, ore, and other materials heavier than grain. Designers followed grain elevator practice as to belts and idlers. One of the

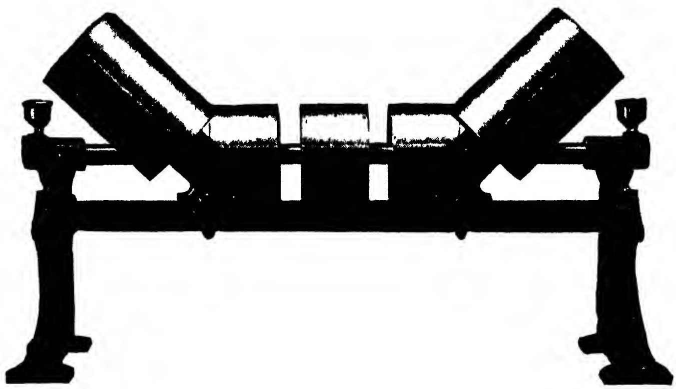


FIG. 2–9. Concentrators Mounted on a Stand with Carrying Pulleys.

largest installations in the early nineties was the ore-concentrating plant of the New Jersey and Pennsylvania Concentrating Company at Edison, N. J. It had over 50 belt conveyors, ranging from 20 inches to 30 inches wide and up to 500 feet long. The first idlers used there about 1891 were heavy cast-iron spool idlers, but these were failures

because, in order to get the greatest capacity from the belts, the troughing was deep and the spools were much larger at the ends than at the middle. These were heavier than the spools used on grain conveyors; the work was continuous, the place dusty, and the belts wore out rapidly. When the plant was rebuilt in 1893, idlers of the type shown in Fig. 2–10 replaced the spool idlers; the new idlers were practically the same as the flat rolls with concentrators used on grain conveyors.

The chief trouble at Edison and at other places doing similar work was with the belts. The sharp pieces of heavy ore cut the fabric and the stitching of canvas belts and loosened the plies; the same thing happened with rubber belts. We know now that much of the difficulty

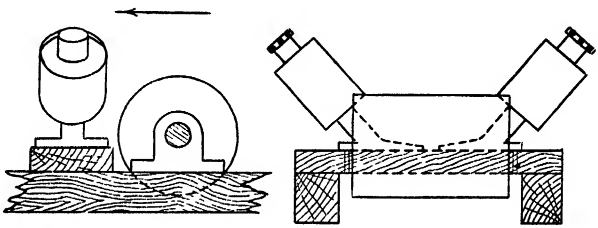


FIG. 2–10. Troughing Idler for Ore Conveyor. (Edison, N. J., 1893.)

was due to improper delivery of material to the belts and to improper construction of the belts themselves. Fig. 2–11 shows a transfer between two belts at Edison; apparently no attempt was made to deliver material to the receiving belt with some velocity in the direction of travel.

Improvements by Thomas Robins. Thomas Robins, Jr., then in the business of making rubber belts, visited the plant about 1891 and noticed the following points: (1) that the thin layer of rubber which covered the belt resisted abrasion much longer than the duck which formed the body of the belt; (2) that each layer of duck wore out faster than the one preceding it, showing that as the belt wore thin and the tension on the threads increased they were cut more easily; (3) that belts wore out in the middle and split longitudinally while the edges were still good (*Trans. A.I.M.E.*, April, 1896). From these observations Mr. Robins concluded that the sole function of the fabric in a rubber belt should be to give the belt tensile strength and that it should be protected from injury by a cover of rubber compound that would resist abrasion better than the cotton threads or the thin layer of friction rubber that covered them. After experiments with various rubber compounds, he furnished belts

with a thick face of rubber that proved to be much more durable than those previously used, lasting years where others lasted months. In 1893 he patented the belt shown in Fig. 2–12, which has an extra thickness of rubber at the center to resist the abrasion that comes from feeding material from a comparatively narrow chute; it resisted

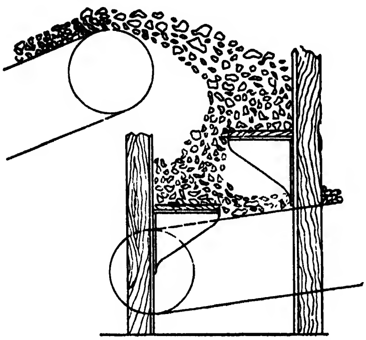


FIG. 2–11. Transfer of Ore between Two Belts. (Edison, N J., 1893.)

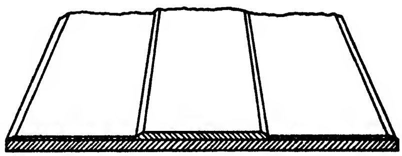


FIG. 2–12. Original Robins' Belt with Extra Thickness of Rubber in Center, 1893.

abrasion very well, but the belt was so stiff that it would not conform to the shape of the troughing idlers, which, as shown in Fig. 2–10, had the side pulleys inclined at 45°.

In 1896 Mr. Robins patented his "stepped-ply" belt (Fig. 2–13) in which the belt thickness is the same for the entire width but the plies of fabric are fewer at the middle than at the edges. The space

FIG. 2–13. Robins' Stepped-ply Belt, 1896.

left by the omission of the plies was filled with the rubber of the top cover, making the extra thickness of protective cover there equivalent to that shown in Fig. 2–12, but using less rubber where the abrasion was not so great. The omission of the plies at the middle made the belt more flexible in cross section, allowed it to conform more closely to the shape of the troughing idlers, and at the same time left a thick stiff edge to bear against the side guide pulleys (*P′*, Fig. 2–14). These pulleys were necessary to keep the belt straight on 45° troughing idlers, especially when the belts were comparatively narrow and stiff and when the inclined idlers came every 4 or 5 feet. Idlers inclined at 45° were in successful use on grain conveyors, but there the belts were relatively wide and thin and the troughing occurred only at the load-

ing points or at intervals not closer than 8 or 10 feet. Under these conditions the grain belt had a good guiding contact with the horizontal pulleys and did not usually require side-guide idlers to keep it straight; but with stiff and narrow belts on 45° idlers, the guiding contact on the horizontal pulleys was slight, the side-guide pulleys were necessary, and the belt had to have a thick stiff edge; otherwise the edges were chafed, bent over, or rubbed off and the belt destroyed.

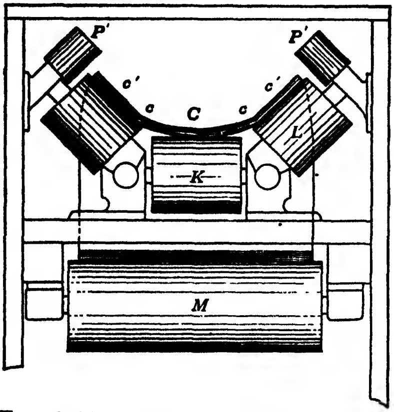


FIG. 2–14. Stepped-ply Belt on 45° Troughing Idlers with Side-guide Pulleys, 1896.

In the early days of the belt-conveyor business, say before 1900, comparatively few belts were wider than 24 inches. Troughing was considered a means to permit a narrow belt to carry what would otherwise require a wider and more costly belt, and the tendency was to use belts narrower than would now be considered good practice.

The Robins patent (571,604, Nov. 17, 1896) covers also the idler construction shown in Fig. 2–15, with the following claims: "Claim 5. The supporting pulleys, *L, K, L,* the hollow bearings *F,* therefor, and the horizontal and turn-up hollow-shafts secured in the said bearings, and the oil devices mounted on the ends of the turn-up shafts, substantially as set forth." "Claim 6. In combination the two brackets or castings suitably supported, the horizontal pulley mounted between them, the turn-up shafts secured in the said brackets or castings and the pulleys *L* loosely turning thereon, substantially as set forth." This idler followed earlier designs in having the troughing pulleys inclined at 45° (*Trans. A.I.M.E.*, April, 1896), but it differed from them in combining the troughing idler group into one self-contained unit by using only two castings to support the pulley shafts instead of four (compare Figs. 2–15 and 2–10). This arrangement with pulleys turning loose on hollow shafts permitted lubrication from two points instead

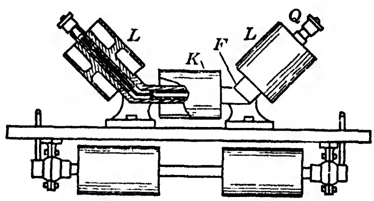


FIG. 2–15. Robins' Three-pulley Idler, 1896.

of four, and at the same time the idler itself was easier to install and to use; there was no distinction between front and back as in the older form where the belt ran on to the horizontal pulley first (see Fig. 2–10). Although the figure in the patent shows the three pulleys in a single plane, that feature is not referred to in the language or the claims of the patent; it seems to have been adopted as a means of effecting lubrication in a certain way.

Prior to 1890 much had been done toward the improvement and standardization of grain conveyors by John T. Moulton, James A. Macdonald, the Webster & Comstock Manufacturing Company, and others interested in the grain elevator business, but belt conveyors had not been used extensively for materials other than grain. Mr. Robins' great contribution to the industry was an active company which began in 1896 to design and sell belt conveyors for handling coal, ores, stone, and all similar materials. Before that time most of the business in this line lay between the belt maker and the pulley maker, neither interested in the other's product and neither capable of giving engineering advice or of insuring definite results from the operation of the conveyor. There was a great expansion in the use of all kinds of conveying machinery between 1895 and 1910, and the belt conveyor had its share of the growth. To Mr. Robins and his company must be given the credit for most of the pioneer work in extending the use of belt conveyors to the handling of materials other than grain and in laying the foundations for the engineering knowledge of the business. Others entered the field, some with devices intended to evade the Robins patents, others with improvements suggested by the growing experience of makers and users of belt conveyors.

Changes in Belt and Idler Construction. The earliest Robins idlers had a rather wide gap between the adjacent edges of the pulleys, a distance sufficient to allow a heavily loaded belt or an old limber belt to sag between the pulleys, as shown in Fig. 2–16. The same defect existed in various two-pulley (Fig. 1-6), four-pulley (Fig. 1-8), and two-plane three-pulley idlers, in which the edges were not brought close together. After the first few years of experience, the gap was made less, and in the majority of modern idlers, the distance between the pulleys is made as small as practicable, i.e., about $\frac{1}{4}$ or $\frac{1}{8}$ inch.

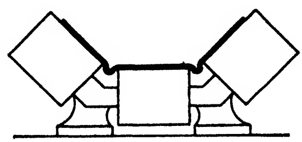


FIG. 2–16. Sag of Belt into Gap between Idler Pulleys.

Two-pulley and four-pulley troughing idlers are now practically obsolete.

When the belt sags between the pulleys of a troughing idler of the kind shown in Fig. 2–16, not only is it in danger of cracking by direct flexure under the load, but also there is a tendency of the edges of the pulleys to seize the sag of the belt and squeeze it together to form a sharp bend. Comparing Figs. 2–17 and 2–18, it is evident that, the greater the angle at which the pulleys are inclined to each other, the greater is the wedge angle toward which the belt travels as it approaches the idler, and the greater the chance that the converging edges of the pulley rims will seize the sag of belt and squeeze it. This squeezing action causes the belt to crack or to split lengthwise.

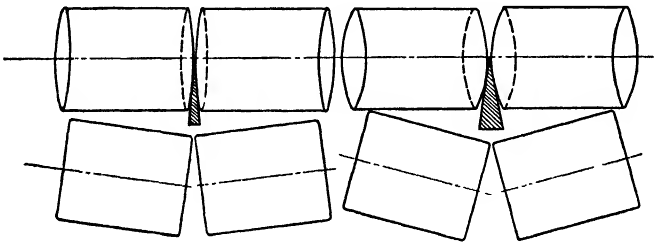


FIG. 2–17. FIG. 2–18.

FIG. 2–17. Angle between Pulleys 15° Small Wedge Angle.
FIG. 2–18. Angle between Pulleys 30° Large Wedge Angle and Greater Liability to Pinch the Belt

From examinations of spoiled and discarded belts, it was noticed that splitting was more likely to occur if the longitudinal joints or seams in one or more plies of duck came just at the point where the belt was bent by the troughing pulleys, or if that line of flexure in a stepped-ply belt happened to come where the thickness of the plies changed, as at *c*, Fig. 2-14. In early days belt makers were not always particular about how the duck was cut and assembled. If the width was made up of several narrow strips of duck and if the butt joints where the edges of the strips met came at the same place in the width of the belt, the result was a plane of weakness which, if it came just at the place where the belt bent for troughing, would lead to the destruction of the belt. As a result of knowledge on this point, belts now made by careful manufacturers have the longitudinal seams or joints at the middle of the belt or near its edges where they will not be affected by the bending of the belt over troughing idlers of the types in general use.

The connection between the steep angle of troughing and the splitting of belts was quite apparent, and since the splitting was a serious difficulty, it was not long before the angle was changed by Robins and others from 45° to 35°, while later idlers were made with the angle 30°, 25°, 20°, and even less. These changes helped belts in still another way, i.e., by reducing the internal stresses in the belt due to the shift of the load in passing over the idlers. In Fig. 2–19, *A* represents the edge of the load on a belt troughed at 45°. Midway between idlers, the edge of the belt drops and the load slips to a line *B*; this readjustment of the load extends in a smaller degree to the material back from the edge of the belt; some of it is on the surface where it can be seen, but some of it is within the cross section where it is not noticed. The action of the inclined pulleys in pushing back the load is exerted through the belt, at the bend it produces a tensile stress on the filler (crosswise) threads in the fabric, and naturally, the less the angle of troughing the less the disturbance of the load and the less the stress in the crosswise threads in the belt.

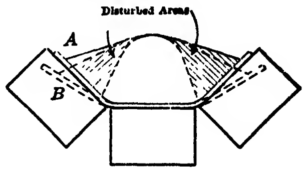


FIG. 2–19. Disturbance of Load Cross Section on 45° Troughing Idlers.

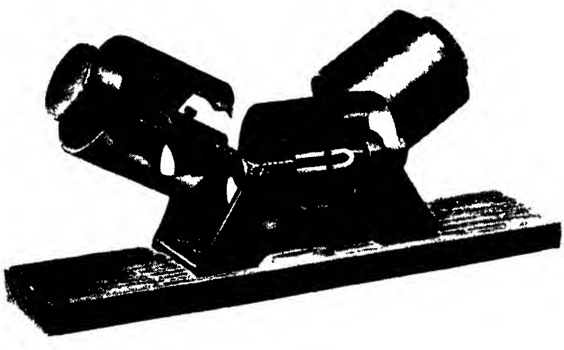


FIG. 2–20. Three-pulley Two-plane Troughing Idler.

On the loss of power due to change of cross section, see page 121.

Fig. 2–20 shows a form of idler free from the defects of earlier constructions. There were no gaps between the pulleys into which a belt might sag and be pinched. That was because the corners of the pulleys overlap as regards the run of the belt. In the Zieber idler, Fig. 2–21, the horizontal pulley has twice as much contact with the belt as each inclined pulley, and the inclination of the two

pulleys is adjustable to three angles, 10°, 15°, or 20°. This construction has several advantages. First, the belt will run straighter because of the broad horizontal pulley. Second, since the horizontal pulley is tight on the shaft, most of the weight of belt and load is carried to closed-end babbitted bearings and not to the bores of pulleys running loose on a shaft. Third, the adjustment of the inclined pulleys

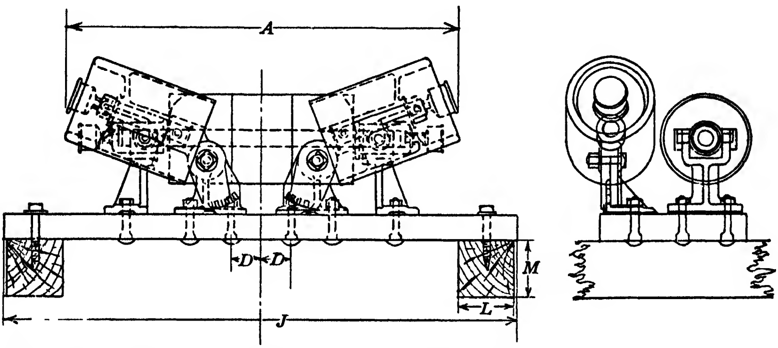


Fig. 2–21. Three-pulley Two-plane Troughing Idler with Broad Horizontal Pulley.

permits the belt to be troughed only as much as is necessary to prevent spill at the loading point and keep material on the belt. Many belts carry loads which do not shift on the belt after they are up to belt speed. The belt can then be troughed as much as necessary to prevent spill or scatter at the loading point, and run with less troughing for the rest of the distance. In general, the flatter the troughing, the better for the belt; it will run straighter and last longer.

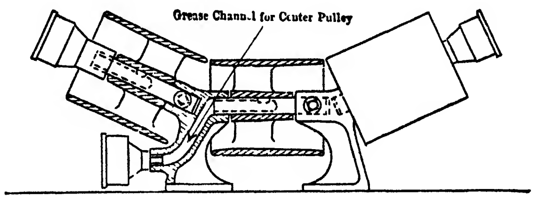


Fig. 2–22. Three-pulley Troughing Idler with Independent Lubrication for Center Pulley. (Link-Belt Company.)

This type of idler is well suited to stitched canvas belts and balata belts; they are stiffer than rubber belts and do not trough so well.

Single-plane three-pulley idlers originally made in the style shown in Fig. 2–15 later assumed the form shown in Fig. 2–22. The pulleys were set with the edges of the rims close together and they were gen-

erally lubricated from two grease cups, but in some designs the lubrication of the middle pulley was made more certain by having a separate grease cup for it.

In the Howard idler, Fig. 2–23, most of the weight of belt and load is carried on a broad center pulley or pulleys fixed to a shaft which runs in babbitted trunnion bearings, each with its own grease cup. The merit of this construction is that the running friction is on babbitted journals and not on a comparatively short hub of a cast-iron pulley with lubrication that is sometimes uncertain. The inclined pulleys act on about one-fourth of the width of the belt at each side and are mounted so that their rims at the bending corner come below the rim of the horizontal pulley; hence the belt cannot be pinched there.

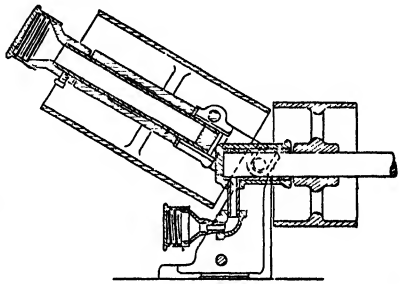


FIG. 2–23. Three-pulley Troughing Idler with Center Pulley Tight on Shaft and Corner of Inclined Pulley Depressed.

Belts Running Crooked. It was a common experience with old-time belt conveyors that the belt would not conform well to the contour of the 45° and 35° troughing idlers, but would tend to lift off the horizontal pulley when lightly loaded or empty. In doing this the belt was likely to be cut by the edges of loading chutes and skirt boards, and, by losing the guiding effect of the horizontal pulley, the belt would run crooked. As has been stated, side-guide idlers were necessary to keep the belts centered on these idlers, and belts were often damaged by the constant pressure against their edges. A later chapter explains why the tendency to run crooked decreases as the angle of troughing is made less. This connection was not generally recognized in practice, but it is a fact that, when idlers with angles of 30° and less came into use, the troubles with belts running off center became less, there was less need for side-guide idlers and it was found that belts of uniform cross section (straight-ply belts) were as satisfactory as those in which greater flexibility and better contact with the horizontal pulley were attained by omitting plies of fabric near the middle of the belt (stepped-ply belts).

Multiple-pulley Idlers. Quite early in the belt-conveyor business there were suggestions that, if idlers were made with more than three pulleys set to maintain the same depth of troughing as with three-pulley idlers, the bending action on the belt would be easier

and the conveyor would suffer no loss of carrying capacity. By having a greater number of pulleys, the belt would be bent through a smaller angle along each line of longitudinal flexure, the internal stress on the filler threads of the belt fabric would be diminished, there would be less risk of pinching the belt between pulleys, and the tendency to crack or split the belt would be less. The complication of parts and the increased difficulty of lubricating the pulleys were obvious disadvantages which delayed the introduction of multiple-pulley idlers; they did not come into use until after 1905.

The first United States patent on a multiple-pulley idler was that granted to Lynch (1899), which shows an impractical device consisting of six pulleys mounted on a bent shaft. Plummer in 1903 patented a five-pulley idler like present ones except that the faces of the pulleys were concaved to match the curve of the belt and avoid all angular bends. Neither of these schemes ever came into practical use.

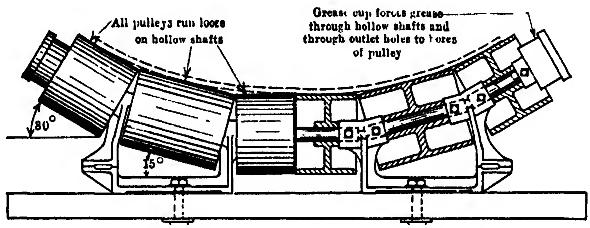


FIG. 2–24 Robins' Five-pulley Idler, 1909.

The four-pulley idler formerly shown in makers' catalogs (Fig. 1-8) was open to the objection that the center of the belt, which is under the greatest depth of load, was likely to sag between the middle pulleys, be rolled into the gap, and split along its whole length. The Robins idler of 1909 (Fig. 2–24) and the Peck idler of 1913 were alike in using five pulleys with the inclined pulleys set at 15° and 30° from the horizontal. Most of the multiple-pulley idlers with grease lubrication formerly used had this arrangement of five pulleys. They differed chiefly in the ways of making and assembling the cast-iron brackets that support the pulley shafts.

Expedients to Avoid Cracking of Belts. The splitting of belts by longitudinal cracks was a serious matter in the early days of the belt conveyor business, and a number of schemes were brought forward to prevent it. Some of these referred to belt construction, others to idler construction. Some have been referred to on previous pages. Of those named below, not one is in commercial use today; but they

are frequently mentioned, and occasionally some similar devices are newly invented and brought to the notice of those in the business.

Belt Devices. The Selleck patent of 1902 covers a conveyor belt composed of a central strip and two side sections flexibly united at their adjoining edges by interlocked lacings, the lacings being coils of wire enclosing a flexible thong of leather. In other words, the inventor starts with a split belt to prevent splitting. Ridgway, in his patent of 1902, omitted some of the plies at the bending points of a rubber belt and increased the rubber there, to get greater flexibility. Some belts of this "hinged-edge" design were used in this country and in England; they troughed well and ran straight, and some gave satisfaction. Other belts showed structural weakness along the lines of bend owing to the comparatively few filler (crosswise) threads of fabric there. In other words, these belts were half split to begin with; the bending was concentrated along the weak lines, and the extra rubber could not hold the belt together. In the three Plummer patents of 1903 the body of a canvas belt is made thick and impregnated and hardened by saturation with drying oils, while the wings are made of fewer plies, and being treated with non-drying oils, remain flexible. The Ridgway patent of 1904 discloses a double belt structure in which the load is carried on a troughed belt which is supported by and receives its trough shape from saddle-shaped cleats mounted on a flat belt running underneath over its own separate pulleys. This scheme was costly and complicated and did not get beyond the stage of advertising and demonstration. Plummer, in 1906, patented a canvas belt in which the upper plies, which take the abrasion of the material carried, are stitched together for their full width, but fastened to the lower plies at their longitudinal central portions only, in such a way as to make the belt flexible for troughing. This belt never came into practical use.

Idler Devices. The Mann and Neemes idler of 1902 consisted of a spool idler made in five or more sections, the middle section tight on the straight through-shaft, the outer ones loose. It was intended to give the troughing of a deep spool idler without the disadvantage of wear on the under side of the belt (see page 25). It would do this if each loose section of the pulley were sure to turn freely on the shaft, but since lubrication through a long hole with many side outlets is uncertain, it is safe to say that some of the pulleys would not turn freely and the belt would rub and run crooked. Besides that, the idler is likely to be expensive if well made. The Rouse idler of 1907 represents another effort in the same direction. Patents have been issued at various times since 1907 on "helical" idlers in which the

supporting pulley is made of a length of steel rod or ribbon wound into a helix of cylindrical shape. In the Thomas idler of 1907, the helix forms a tension spring that bends to conform to the weight of the belt and its load, the ends of the rod being extended on the axis of the helix to enter ball-thrust trunnion bearings carried by a stand at each side of the conveyor.* One conveyor equipped with these idlers was set up in Chicago; when a load was put on the moving belt the springs vibrated and set up a wave motion in the belt which made it inoperative. Vrooman, in 1909, patented an idler of similar form with a number of separate pulleys fitted to a flexible-jointed shaft with a spring-mounted end-thrust bearing at each side. This had the defects of the Thomas idler with some added complications. The same may be said of the Proal patent of 1911, in which it was proposed to use separate pulleys joined together by axial helical springs and carried in end-thrust bearings similar to the above.

The old difficulties with the cracking of belts were lessened or avoided not so much by radical changes in belt construction or by novelties in idler construction as by reducing the angle to which the belt is bent along the line of flexure in troughing. This led to the use of three-pulley idlers with the two bends 30°, 20°, and as low as 10°, and to the use of five-pulley idlers where each of the four bends is 15°. The tendency of the present day is in the same direction, toward wide belts, shallow troughing, and simplicity in idler construction.

Efforts to Resist Cutting and Abrasion. Early experiments by Robins showed that rubber specially compounded resisted the action of a sand blast much better than did other materials, the wear being in the following ratios: rubber, 1.0; rolled steel, 1.5; cast iron, 3.5; various cotton belts, 5 to 9. These ratios have been quoted by some as showing the comparative worth of these materials for conveying gritty substances, but the comparisons are not valid when, as in most belt conveyors, the action is not simple abrasion, but a combination of cutting and abrasion. It is well known that, when a belt with a rubber cover is subjected to severe impact of large sharp pieces, the cover may be cut through to the fabric, and dirt and water may get under the cover and into the fabric, whereupon the belt fails by the coming loose of its cover or by the separation of the plies while the cover may still retain its original thickness. The threads of cotton in a rubber belt are to a certain extent waterproofed by the layer of "friction" which covers each ply of duck, but the fibers of the threads are not impregnated with rubber and they still retain capacity for

* *Foerdertechnik* for August 16, 1939, refers to the present-day use of idlers of this kind in Germany.

absorbing moisture. When a cut penetrates to the duck, capillary absorption will spread any water which enters there, and when the cotton decays from mildew the plies separate or the cover comes loose in a blister.

Metal Reinforcements. The steeply inclined belt conveyors known as "tailings stackers," used on gold dredges to carry off the waste, are subject to very severe cutting and abrasion and the action of water. The material runs from fine gravel to rocks weighing 200 or 300 pounds and may include tree stumps, pieces of lumber, and anything else picked up by the dredge buckets. Costly belts used in this service have lasted only a few weeks or months, and since they were often scrapped because of wear in the middle while the edges were still good, efforts have been made to add life to these belts by riveting metal cleats to the working face (St. Clair, 1890; Manning, 1908; Cory & Dandridge, 1909); by metal staples (Folsom, 1906); by flexible woven wire face (Hohl & Schorr, 1906); by imbedded wire coils (Pattee, 1916; Heaton, 1909); by leather face (Cook, 1906). None of these has come into practical use; there are objections to all of them. Wherever a hole is made through the fabric for a metal reinforcement, water and sand will enter; if the reinforcing pieces are arranged in parallel rows, the belt may crack by concentration of the bending along those lines as it runs over the troughing idlers. Wire reinforcement rusts out as well as wears out; the bond between the metal and the rubber or the fabric is generally imperfect, and, as the parts separate under the stresses of troughing and bending over pulleys, water and grit work their way into the belt.

Other Protective Devices. Another protective device is shown in the Vaughan patent of 1905, where the main conveyor belt is partly covered by a separate narrower belt that acts as a protector to the middle of the wider belt. In the Voorhees patent of 1906, a belt has a rubber cover in which are imbedded cotton fibers set to expose their ends to the impact and abrasion of the material carried. Plummer, in 1906, made a compound belt with a solid-woven cotton back for flexibility, with a wear-resisting face of stiff canvas stitched on. In 1916 Bowers patented a rubber belt in which the wearing face contained several edgewise layers of frictioned fabric set on the bias so as to take the wear partly on the ends of the cotton fibers.

Rubber Covers. In the rubber-belt business as in the rubber-tire business, it has been found that nothing protects the fabric carcass of the belt or the tire so well as a rubber cover or tread. Metal reinforcements and other protective devices add expense and generally fail to protect the body of fabric from abrasion, cutting, and the entrance

of water. Rubber covers are depended upon to do that in present-day practice, and in the quality of the covers and their ability to withstand abuse there has been a steady improvement in recent years.

Lubrication of idlers was a troublesome problem in the early days of the business. Oil lubrication (Fig. 2–15) was not a success; oil dripping or scattering from the pulleys got on the belt and damaged it. Grease was found to be better. The pulleys did not turn quite so freely, but the leakage of grease from the bores of the pulleys formed a collar which helped to prevent dirt and water from getting in. Grease usually did not drip on the belt and its use was an advantage in places like stone-crushing plants where the air was full of dust.

FIG. 2–25 Rim of Idler Roll Worn away by Belt Sliding over it.

In all grease-lubricated idlers, screw-cap grease cups were depended on to force the grease through axial holes into the bores of the pulleys (Fig. 2–20). If the grease hardened in the holes, or if the holes were too long, or if the threads on the cap of the grease cup were too loose a fit, grease could not be forced through the holes, the pulleys would run dry, then wear out. See Fig. 2–25. Figs. 2–21, 2–22, 2–23 show how some designers tried to avoid this trouble. In five-pulley idlers it was possible to reduce the points of lubrication to two grease cups at the outer ends of the outer shafts, Fig. 2–25. There was an advantage in having the cups within easy reach of the man who kept the idlers lubricated, but getting grease into the bores of five separate pulleys was even more uncertain than in three-pulley idlers where there are only three side outlet holes and where the axial grease passages are relatively shorter. For this reason some makers provided their five-pulley idlers with four or even five cups. See Fig. 2–26.

Fig. 2–26 shows the idea back of the five-pulley idler, that is, that the faces of the pulleys conform to the natural curve of the belt without decided bends, and that the belt was not likely to be creased or cracked. To get that result and avoid gaps between the rims

FIG. 2–26. Natural Sag of Belt on Five-pulley Troughing Idler.

of the pulleys, the hubs of the pulleys had to be made short enough to leave room for the tops of the cast-iron stands which take the shafts. Usually, the length of the hub was less than half the face of the pulley, and as a result, especially with poor lubrication, the pulleys were likely to wear loose, run eccentric, and rattle.

FIG. 2–27. 24-inch Belt on Cast-iron Spool Idlers Carrying Cement Clinker (1920).

Spool or Flared Idlers. The difficulties and uncertainties of forcing grease through hollow shafts into the bores of three or five cast-iron pulleys caused some users of belt conveyors to prefer the old-style

spool idler, Fig. 1–4 and Fig. 2–4, even though it did chafe the belt on the under side. Fig. 2–27 shows a clinker conveyor in a Pennsylvania cement mill; the idlers had a cast-iron center pulley with two cast-iron bell-shaped ends which lifted the edge of the belt an inch or two, all tight on a shaft which revolved in two babbitted bearings, each with a grease cup. These idlers were simple and strong and could withstand abuse and neglect. The belt ran nearly flat, the slight lift of the edges prevented spill, but the carrying capacity was only about half as much as that of a belt on troughing idlers.

For sheet-steel idlers of this form, see page 115.

Roller-bearing Idlers. About 1915, Hyatt roller bearings (Fig. 2–28) were applied to five-pulley idlers in the manner shown in Fig. 2–29. Each pulley was bored to receive the hardened steel sleeve which forms the outer shell of the bearing; steel plates pressed into the bored hole held the roller bearing in place, and loose steel washers filling the space between the pulley hub and the hub of the stand took the end-thrust due to the inclined position of the pulley. Four grease cups were used for each troughing idler, so that grease did not have to be forced through the roller bearing in one pulley to reach the next one.

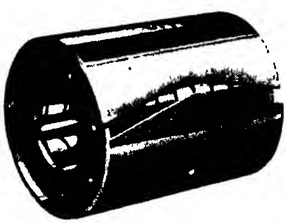


FIG. 2-28. Hyatt Roller Bearing.

Another consideration which has led to the disuse of idlers with

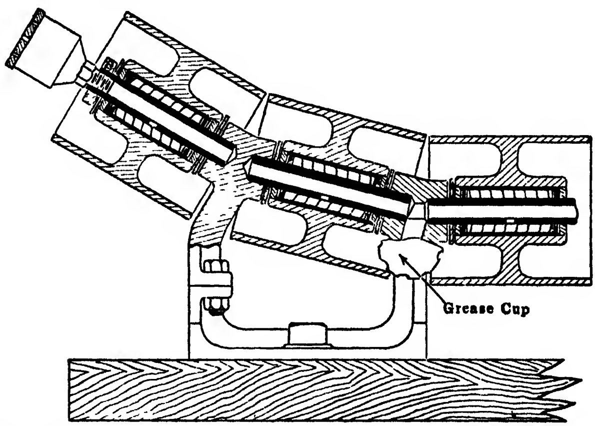


FIG. 2-29. Roller Bearings Applied to Five-pulley Troughing Idler (1915).

plain bore pulleys is the large amount of grease required to lubricate them. The following is taken from the second edition of this book:

In some large plants it requires the full time of several men to keep grease cups filled and the caps screwed down. It sometimes happens that the first evidence of lack of such care is costly damage to the belt. If the attendant misses a cup, no grease is forced through the hollow shaft, a pulley refuses to turn, the belt slides over it, wears the rim thin, and finally cuts it away with the result that the belt is cut on the sharp edges. This is more likely to happen on the return run where the grease cups are low and hard to get at and where the idlers are often concealed by the supporting frame or by the protective deck. In this location a failure of the pulleys to turn is not so easily seen as when that trouble occurs on the carrying run.

In some plants equipped with many belt conveyors the cost of grease and the labor of filling and adjusting the cups amounts to a large sum in the course of a year. At one plant that uses about 15,000 feet of wide belts, grease costs over $2000 a year and the labor charge is over $7000 a year (1920). All the belts in this plant do not run every day; if they did, the charges would be considerably more. In another plant that uses about 8000 feet of belt four men are employed in attending to grease cups on the conveyors.

In some plants the expense of lubricating, repairing, and replacing grease-cup idlers amounts to more than the cost of belt renewals.

In plants with only one or two belt conveyors the machine is looked after with care and perhaps some pride as well; then the ordinary grease-cup idlers may receive the attention they deserve. But in a large plant with many conveyors the identity and importance of separate conveyors are lost. To fill and screw up grease cups regularly and faithfully in such a place calls for the steady work of one or more men, and the attention which these men give to their work greatly influences the cost of repairs, the cost of shutdowns, and the cost of carrying a ton of material. A routine job of that kind is not easy to supervise and consequently it is often shirked.

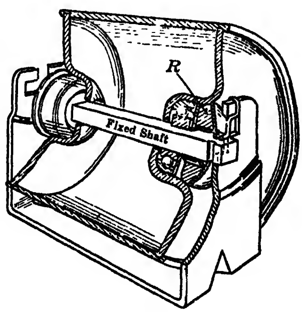


Fig. 2–30. Stephens-Adamson Ball-bearing Pulley.

The Unit Carrier, Fig. 2–30, of the Stephens-Adamson Company (Aurora, Ill.), first made in 1911, represents an early development of a ball-bearing idler into commercial form. With improvements suggested by years of use, it is still on the market. The "unit" consists of a sheet-steel roller 6 inches in diameter by 6 or 8 inches face, two cartridge-type ball bearings which are renewable, a ½-inch-square shaft, and a one-piece sheet steel supporting bracket. The units can be assembled on steel channels into groups of three or more with the outside rolls set at 20° angle

to trough belts of any width. The ball bearings are packed with grease at the factory and under favorable conditions will run for a long time. To renew the lubricant the bearings are removed, cleaned, repacked, and replaced.

The Stearns idler made first (1921) by the Stearns Conveyor Company, Cleveland, Ohio, was composed of a number of sheet-steel rolls of the "unit" type fitted with roller bearings mounted in a central tube and running on a hollow shaft. Fig. 2–31. The shaft had a fitting at one end to take the nozzle of a grease gun. The method of lubricating by a pressure gun was original with Stearns; it was successful and is now used on all forms of belt idlers in this country and abroad.

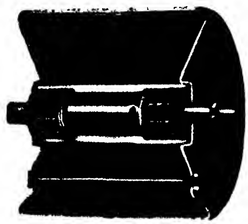


FIG. 2–31. Stearns Unit Pulley of 1921.

Between 1915 and 1920, tests of belt conveyors equipped with ball-bearing or roller-bearing idlers showed that they required considerably less power than if the idlers had been of the old style with plain bearings and grease-cup lubrication. Table 2–A shows a number of comparisons.

TABLE 2–A

Belt Width	Centers	Lift	Tons per Hour	Speed	Idlers	Actual Test Horsepower	Estimated Horsepower for Plain Idlers
48 in.	387	109 ft.	795	483	Hyatt	107	137
48 "	405	119 "	1030	506	"	162	190
42 "	600	114 "	324	242	"	52	71
44 "	675	. . .	540	358	Ball bearing	20	56
42 "	605	76 "	1220	?	Hyatt	134	198

In one case cited by the makers of Hyatt roller bearings the reduction in the horsepower required made a saving of $1000 in the price of the motor, and since a belt of fewer plies could safely be used, a reduction of $2700 was made in the cost of that item.

Advantages of Ball-bearing and Roller-bearing Idlers. As compared with old-style idlers, idlers with ball or roller bearings have several advantages.

1. Less power to run the conveyor.
2. Less tension on the conveyor belt.

3. Less expense for attendance and for lubricant.
4. Less chance that the idlers will stick tight or drag and injure the belt.

5. **Saving in Repairs and Replacements.** The freedom with which ball-bearing or roller-bearing idlers turn means more than a saving of power; it means that there will be less rubbing on the rims of the pulleys, fewer replacements of pulleys, and less chance of cutting and scraping the belt over pulley rims that have been worn thin and cut through. Replacement of idlers is not a serious matter in most plants, but in others, even where gritty material is not handled, grease-cup idlers wear out regularly. In one plant that uses about 8000 feet of belt for conveying run-of-mine and crushed coal, the replacements of idlers for worn-out and broken pulleys and stands per year for a number of years averaged over 15 per cent of the number in use.

6. **Saving in Labor and Attendance.** They save also in attendance and cost less for grease. It is estimated that for one year of ordinary service the following quantities of grease are required for each cup of grease-lubricated idlers: No. 2 cup, 6 pounds; No. 3 cup, 12 pounds; No. 4 cup, 20 pounds. The time needed to fill each cup, over and above the ordinary screwing down, will average about 2 hours per year per cup.

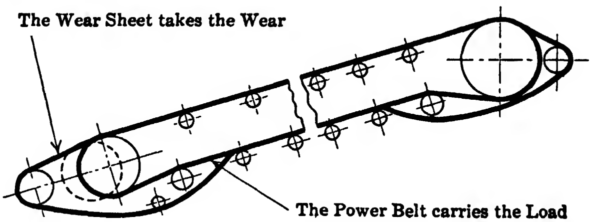


FIG. 2–32. "Duplex" Conveyor with Two Belts.

Unusual Belt Conveyors. The *Duplex system* is a trade name given to belt conveyors of the type shown in the Kyle and Mitchell patent 1,626,041 of 1927, in which one belt called the "wear sheet" lies on and over another called the "power belt" and moves with it; see Fig. 2–32. The "wear sheet" is composed of a few plies of fabric plus a good rubber cover, and when it wears out, it can be renewed without disturbing the power belt, which may be a transmission belt without any cover. A number of such combinations have been built, and some are successfully handling sharp stone, coke, and similar abrasive materials. The conveyor is made by the Boston Woven Hose and Rubber Company, Cambridge, Mass.

Another scheme for separating the wear belt from the power belt is shown in the Crossen patent 1,970,842 of 1934. It has not been used.

The Lemoine patent 1,405,233 of 1922 proposed setting two belts edge to edge to form a V-shaped trough. In 1926 a car unloader was patented, 1,600,383, by Ahlskog, in which three belts formed the bottom and sides of a moving rectangular trough. None of these came into practical use.

Combinations of Belts and Wire Ropes. When the art of belt conveying was young, minds of inventors were troubled by thoughts of the "injuriously high stresses" in the belts and the "prohibitive cost of the supporting idlers."

Catlin in his patent 826,312 of 1906 proposed to avoid these difficulties by "transferring the stresses of load and motion ordinarily borne by the belt to cables which both support and propel the belt." Warner in patent 1,530,707 of 1925 shows an improved device of that kind, and Gammeter in 1929 discloses in patent 1,726,555 the application of the idea to build conveyors of very great length. The belt does not rest on idlers, but is supported in troughed form on a set of parallel moving wire ropes. The ropes take the tensile pull; the belt supports the load of material. One machinery group may drive some hundreds of feet of wire rope; when several groups of machinery are arranged in tandem fashion, and one continuous belt lies on and over the several sets of ropes, bridging the gaps between one set of ropes and the next set, transfers of material are avoided and the conveyor may be of very great length. This device had not gone beyond the patent stage in 1940; modern practice tends toward reducing idler friction and making belts so much better and so much stronger that they will do the work of conveying material over distances of thousands of feet, and do it without the aid of auxiliaries and what may be called "stunt" devices.

The Johns Conveyor is strictly not a belt, but a moving rubber pipe which encloses material and conveys it in any direction. The pipe is formed by two lengthwise halves which are molded with tongues and grooves on the edges so that a longitudinal twist will hold them together. It is made by the Osborn Manufacturing Company, Cleveland, Ohio, in sizes 2, 3, or 4 inches inside diameter and is covered by several United States patents of which 2,013,242, dated September 3, 1935, is one. At the loading point the halves are separated so that material can be fed into the interior of the pipe; at the discharge point the halves are held apart so that the material falls out. Fig. 2-33, taken from the patent mentioned, shows the cross section of the pipe; a coil chain molded within the rubber of each half gives the pipe

its tensile strength. The rubber pipe is driven by a sheave at one point and is said to require but little power. It may be led by guide

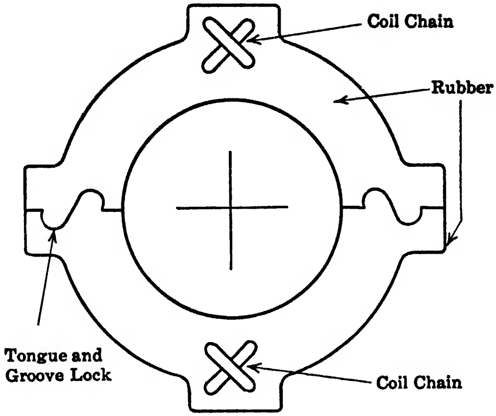


FIG. 2-33. Cross Section of the Johns Conveyor. (Osborn Manufacturing Company.)

wheels into various paths and will convey material in several directions without transfer. Since the driving sheave acts on the soft rubber surrounding the chain and not directly on the chain, the pull in the

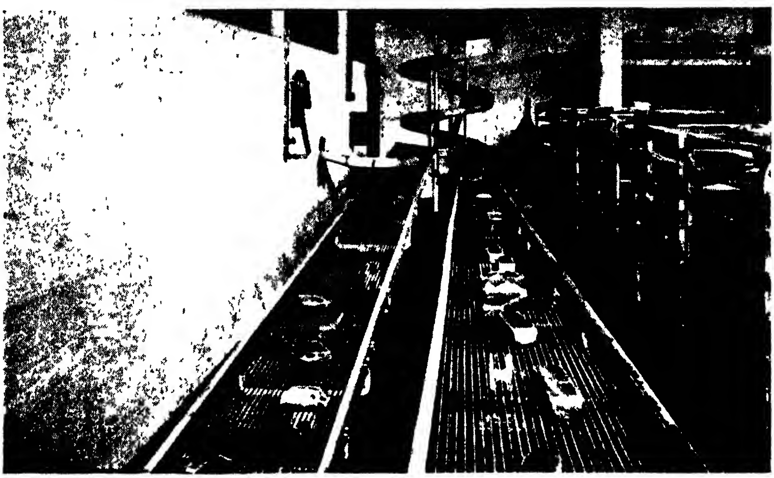


FIG. 2-34. Two Conveyors Each with a ¼-inch Endless Wire Rope. (Link-Belt Company.)

conveyor is limited to what tension may safely be transmitted from the rubber to the chain.

Steel Rope Belts. A few steel rope belts have been used in the United States. A flexible steel rope spliced endless is laced back and forth between a grooved driving drum and a grooved foot drum with one bend over a sheave which acts as a tightener and also leads the rope from the top of one outside groove on the foot drum to the bottom of the outside groove at the opposite end of the drum. As shown in Fig. 2-34, it was used to carry and cool freshly baked bread. The necessity for resplicing the rope occasionally was an inconvenience, because few mechanics can do the job right. In later conveyors built for the same purpose multiple strands of light malleable-iron chains have been used instead of the wire rope.

CHAPTER 3

BELTS AND BELT MANUFACTURE

RUBBER BELTS

The Duck. The strength of the belt lies in the duck, a cotton fabric which for conveyor and elevator belts differs from ordinary sail duck or canvas in the fact that the strength of the warp (lengthwise) threads is considerably greater than that of the weft or filler (crosswise) threads. Duck for rubber belt is graded as 28-ounce, 36-ounce, etc., according to the weight of a piece 36 inches long in the warp by 42 inches wide; the warp threads are usually larger and closer together than the filler threads, and the size of the thread is determined by the number of spun yarns it is made of. Thus, one weave of 28-ounce belt duck may have a warp 6 yarns to the thread, 26 threads per inch, and a filler 5 yarns per thread, 17 threads per inch. Another belt maker may use a 28-ounce duck with a warp 6×24 and a filler 5×14, but with a tighter twist in the threads to make up the weight. The strength of the duck depends not only on its weight, but also on the degree of twist in the threads and the length of the cotton fiber. The cementing action of the rubber depends upon the openness of the weave as well as on the quality of the rubber. All these factors are important; in the best use of them the belt maker shows his knowledge and skill.

Duck is graded also according to looseness and tightness of weave as soft or hard. Soft ducks are more open in structure, have larger voids between the threads, and make a flexible belt. Hard or "silver" ducks are denser and stronger and are used for conveyor and elevator belts subjected to hard work and severe tension.

Friction. The cementing layer of "friction," as it is called in the trade, is a plastic mass of compounded rubber pressed into the layers of duck to hold them together and give the assembled plies which form the "carcass" of the belt the necessary degree of resilience and elasticity combined with resistance to deformation. Frictions are graded as 10 pound, 15 pound, 20 pound, etc., according to the pull required to separate one ply from another at a definite rate in a belt strip 1 inch wide cut from the finished belt. (See page 53.)

To make a batch of friction rubber or rubber for covers, a weighed quantity of crude rubber and weighed quantities of compounding ingredients are put into a mixer and kneaded together for a predetermined time. The plastic mass passes from the mixer to a roller mill, is rolled into strip form, cooled, cut into pieces for easy handling, then stored until wanted. A test piece is cut off to check the quality and composition of the mix.

Manufacture. The duck comes from the textile mill in rolls usually 54 inches wide. It is run over heated cylinders to remove most of its moisture, then between the lower and middle roll of a calender press whose three rolls are heated by steam or cooled by water. The plastic

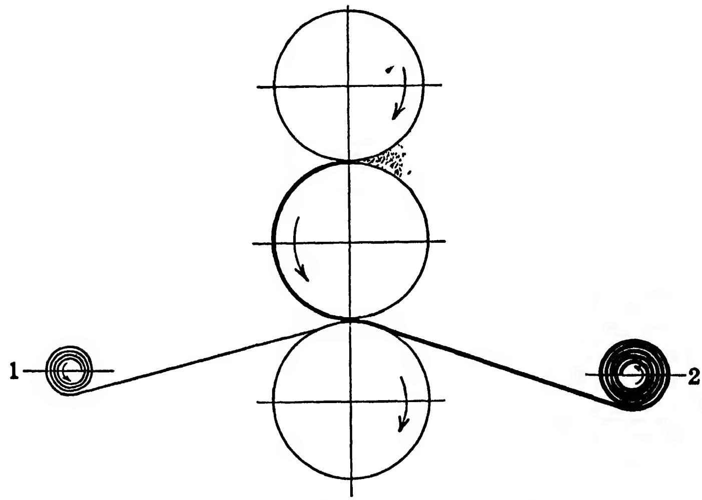


FIG. 3-1. Operation of Calender Rolls.

mass of friction rubber is fed between the top and the middle roll (Fig. 3-1); the middle one is hotter, and the rubber sticks to it; and since this roll turns faster than the lower roll, the rubber is pressed and wiped into the fabric as the duck passes between the two rolls. It is not possible to increase the thickness of the friction layer by decreasing the pressure of the rolls, for then the friction would not be pressed sufficiently into the fabric of the duck and into the voids between the threads. If the duck is a hard duck, it is usual to improve the adhesion of the plies to each other, or the adhesion of the cover to the carcass, by placing between them a "skim coat," that is, a sheet of good rubber rolled to a thickness of about 0.01 inch. When both sides of the duck have been frictioned, it is cut into strips of the desired width and then assembled into belt.

To make a belt of "plied" construction (Fig. 3-2), the plies are each cut or woven to the width of the belt and laid one over the other. The making of a belt of folded construction, say 18 inch 6 ply, Fig. 3-3, starts with an inner strip 36¼ inches wide folded and rolled to make two plies or layers with a butt joint. Then a strip about 36⅝ inches wide is folded around with the joint about 2 inches from one edge,

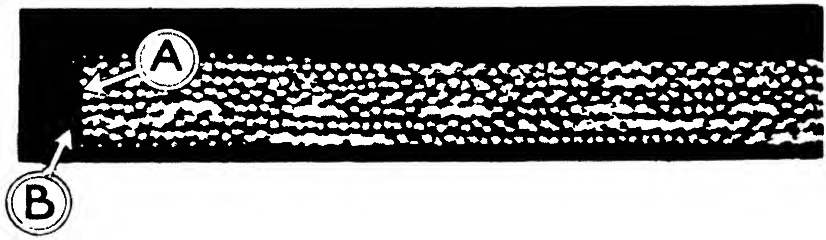


FIG. 3–2. "Plied" Construction of a Rubber Belt.

then rolled, making two more plies; the outer plies are made of a strip 37 inches wide wrapped around the other four, with its joint (which is open about $\frac{1}{16}$ inch) located 2 inches from the other edge of the belt. If the belt is not to receive a rubber cover, the outer joint is closed by a seam strip of soft rubber; then the belt is given a pass through a roller press which squeezes all the plies together and rolls the seam strip into and over the outer joint in the duck.

FIG. 3–3. Six ply Grain Elevator Belt with Friction Surface, Showing Seam Strip.

Rubber Covers. A belt built up of frictioned fabric only is called a "friction surface" belt (Fig. 3-3), but most conveyor and elevator belts are provided with rubber covers. Thin covers may be calendered onto the outside ply; thicker covers are rolled out in sheet, cut to

width, laid over the assembled plies, around the edges, and far enough under the bottom ply to secure a good bond there. Rubber covers are graded as to quality by their tensile strength; see Table 3-B, page 55.

Vulcanizing. At the end of the assembly operations, the belt is soft and "uncured," the rubber in the friction and in the cover is plastic and not elastic, the plies are not firmly cemented together, and the edges are not formed. The belt is finished by the process of vulcanization. This is usually done in a hydraulic press (Fig. 3-4) with upper and lower platens 25 or 30 feet long heated with steam. Steel strips laid on the lower platen confine and mold the edges of the belt. A part of the assembled belt as long as the platens is run into

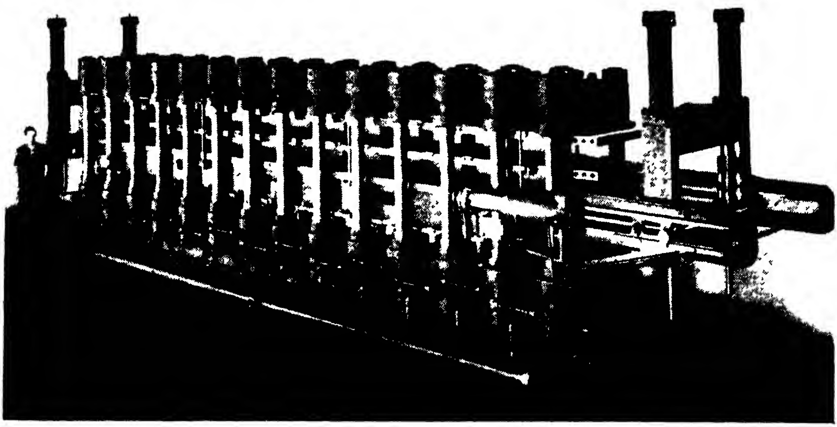


FIG. 3-4. Hydraulic Press Used in Vulcanizing Rubber in Belts. (Farrell-Birmingham Company.)

the press, squeezed at a high pressure and at a temperature of about 280° F. for 30 minutes or less, pulled through, another 25 or 30 feet operated on, and so until all the length of the belt has been vulcanized. After an inspection, samples are taken for testing and the belt is allowed to cool, rolled up, then crated for shipment.

Most rubber belts are put under stretch in the press before the platens close and are cured under stretch; sometimes the duck may be stretched before it is assembled into belt.

Shrinkage of Belts. After they are taken out of the vulcanizing press, rubber belts may shrink slightly and may continue to do so for a month or more. It is customary in belt factories to allow for shrinkage by cutting new belts longer by 6 or 8 inches in 100 feet, but greater

shrinkages than this have occurred and the belts have measured too short when put in place. It is generally possible, however, to apply tension enough to pull such belts to their proper length.

Continuous Vulcanization. The Boston Woven Hose and Rubber Company, Boston, Mass., uses a process and apparatus (Bierer patents and Knowland patents, 1935-6-7) in which the uncured belt is squeezed as it passes around a cylinder heated to vulcanizing temperature, is kept under controlled tension, and comes out of the machine a finished belt. Advantages claimed for belts made in this way are: (1) uniform vulcanization, no "overcure" at press-laps; (2) uniform stretch and elasticity throughout the length of the belt. In Fig. 3-5, *A* is the uncured belt, *B* a set of tension rolls, *C* an endless steel band as wide as the machine, *D* a heated roll which presses the steel band and the belt against the vulcanizing cylinder *E*, *FF* are rolls to guide the steel

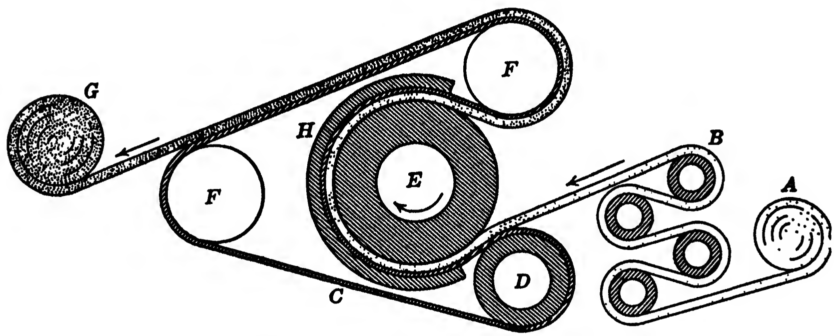


Fig. 3-5. Continuous Vulcanizer.

band *C*, *G* is the cured belt. A steam-heated shoe *H* helps to keep the belt at the vulcanizing temperature.

Technique of Rubber Manufacture. When the latex or liquid rubber is drawn from the tree, it is subject to fermentation and putrefaction as are other vegetable juices. These changes are stopped by smoking the liquid, as is done in the Brazilian forests, or by coagulating it with acid as is done with plantation rubber in the East Indies. Nevertheless, the raw rubber of commerce is to some extent subject to deterioration from the absorption of oxygen from the air, and its physical properties are greatly affected by temperature. Vulcanizing, invented by Charles Goodyear and others about 1840, transforms raw rubber into a product more durable, more elastic, and less subject to change of temperature. Goodyear used sulphur and white lead with heat to bring about vulcanization; since his time there have been many improvements in rubber making and vulcanizing.

Rubber Compounding. Technical works on rubber manufacture give many formulas for friction compounds and cover compounds, but they are of no particular interest to users of belts. It is sufficient to know that hundreds of grades of raw rubber are on the market, and hundreds of compounding ingredients are in regular use. Manufacturers use also several kinds of rubber substitutes made by oxidizing vegetable oils like linseed oil, and many grades of reclaimed rubber are used in the business, some of which cost more than certain grades of raw rubber. A synthetic product "Neoprene" (see page 50) with many of the properties of rubber and some valuable ones of its own is coming into use. Some minerals used in compounding, like whiting, are inert fillers, but others such as zinc oxide toughen rubber and make it stronger. Carbon black has a similar effect. Some ingredients known as accelerators are added to the mix to shorten the time of vulcanization; others increase the range of time and temperature over which correct vulcanization occurs; still others, known as anti-oxidants, give rubber longer life either by hindering the absorption of oxygen from the air or by limiting the changes due to vulcanization to the period during which the belt is in the press.

As a result of the attention which manufacturers of belts have given to rubber compounding, the life and durability of rubber belts, especially in the covers, have steadily improved.

Belt Construction. Most conveyor and elevator belts are of folded ply construction. Manufacturers are generally careful to make widths and folds of the plies in a conveyor belt so that the longitudinal seams do not come at places where the belt bends at the angles between the pulleys of the troughing idler. In Fig. 3-6 there are no longitudinal seams in the inside plies, and those in the outside ply are 3 inches from the edge, where there is little or no bending. In the elevator belt shown in Fig. 3-6 the seams are kept away from the edges and from the middle where the belt is bent longitudinally in passing over the crown or high center of the pulleys. It is bad practice to make up the width of the inside plies of any belt by assembling narrow strips of duck which would otherwise be waste from the slitting operation; such a belt will be sure to crack lengthwise. At one of the western smelters, a heavy 42-inch wide belt failed by longitudinal cracking after a few

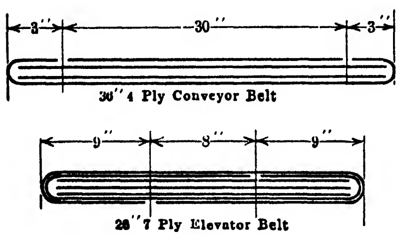


FIG. 3-6. Typical Location of Longitudinal Seams.

months' use. Examination showed that all the inner plies were of the full width of the belt, but the wrapper which formed the two outside plies was butted at a place which happened to come over the gap between two pulleys of the troughing idlers. This gap was $1\frac{1}{8}$ inches wide. The point of weakness in the outside plies localized the bending there; the belt sagged into the wide gap, was jammed between the pulleys, and was ruined.

On the subject of longitudinal joints, the British Standard Specification (see page 54) says: "When there is a longitudinal joint in a ply, the distance from either edge for belting up to and including 20 inches in width shall not be less than one-eighth of the width. For belting over 20 inches, the distance shall not be less than 4 inches. The joints shall be so arranged in the inner plies as to be evenly balanced on either side of the center line of the belting and no two joints in adjacent plies shall coincide."

In making up the length of a belt from shorter lengths of duck it is important to keep the transverse joint in one ply well away from the transverse joint in any other ply; otherwise there may be a plane of weakness across the belt. For similar reason, no transverse joint should be near the point of splice. Requirements of the British Standard Specification are as follows: "Transverse joint should be made at an angle of about 45 degrees; in the outer plies they shall be not less than 250 feet apart; in the inner plies not less than 50 feet apart; no more than two joints in any one ply in 500 feet of length; no joints in adjacent plies closer than 10 feet apart; no joints in any two plies shall coincide."

In the "plied" construction shown in Fig. 3-2 there are no longitudinal seams. It is used for conveyor or elevator belts that have a rubber cover on at least one side sufficiently thick to form a good bond with the rubber strips which seal the cut edges of the plies on each edge of the belt. These rubber strips or edge cushions *A* are generally held to the edges of the plies and to the rubber between the plies by strips of strong open-mesh fabric *B* which at the same time lap over and under the belt.

Stepped-ply Belts. This construction is shown in Fig. 2-13, page 13. Most troughing idlers nowadays have pulleys inclined not over 20°, and straight-ply rubber belts have enough flexibility crosswise to make it unnecessary to omit plies at the middle to make the belt conform to the contour of the idlers. As to the cover, if $\frac{3}{16}$ inch of rubber is necessary in the middle to resist abrasion, it is good insurance and does not cost much more to extend the $\frac{3}{16}$-inch thickness to the edges of the belt. A straight-ply belt is stronger than a step-ply belt of the

same edge thickness; a splice holds better, and there is more strength in the middle to resist the extra tension which comes from running over the crowns of the pulleys. Moreover, there is not the risk that the belt may be split lengthwise as may happen to a step-ply belt at the places where the steps occur.

The belt construction with the added pad of rubber shown in Fig. 2-12, page 13, is still heard of, but in general it is better to buy standard belts and make loading conditions such that there is no excessive abrasion at the middle of the belt.

Resistance of Rubber Covers to Combination of Impact and Abrasion. The protection which a good rubber cover gives to a belt can be

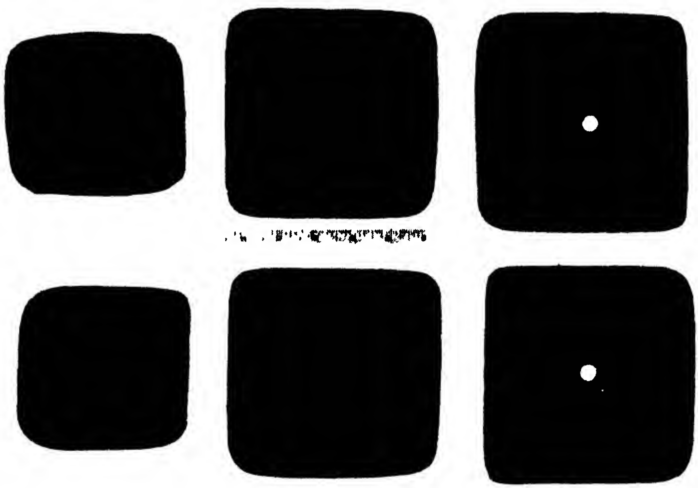


FIG. 3-7. Samples of Belt after 100 Hours of Impact and Abrasion.

seen in Fig. 3-7, which shows the appearance of six samples of belting after a test of 100 hours. Two pieces of 6-ply stitched canvas belt, two pieces of 6-ply high-grade rubber belt with ⅛-inch cover and two pieces of medium-grade belt 6-ply with ⅛-inch cover were tumbled together for 100 hours in a cleaning mill with charges of hard iron castings weighing about 750 pounds per charge. Weights were taken at the time new charges of sandy castings were put into the cleaning mill.

The pieces were all 6 inches square at the start and were cut so that one edge in each specimen was the edge of a conveyor belt. The high-grade belt, distinguished by the hole drilled in the test specimens,

showed least loss of size and weight, and the canvas belt lost most. The pulley side of the rubber belts showed more wear than the other side, but the friction rubber offered considerable resistance to the combined action of the sandy grit and the blows which it received in the cleaning mill. Table 3-A records the weights and losses during the 100 hours.

TABLE 3-A

Comparative Resistance of Belts to Blows and Abrasion

(Specimens 6 inches square. See Fig. 3-7)

Hours of Test	Stitched Canvas Belt		Medium-Grade Rubber Belt		High-Grade Rubber Belt	
	Specimen 1, Weight, grams	Specimen 2, Weight, grams	Specimen 1, Weight, grams	Specimen 2, Weight, grams	Specimen 1, Weight, grams	Specimen 2, Weight, grams
At start	203 5	204	305	304 7	305 8	314
7	201	202	301	300	301	310
14	192	191	295	295	296	306
21	185	186	289	290	292	301
28	170	171	280	281	285	295
35	155	155	269	272	279	286
42	140	140	263	266	271	281
49	133	131	257	258	263	270
56	127	124	254	255	257	267
63	115	115	247	247	251	259
71	108	107	244	245	248	257
79	102	101	240	240	245	254
87	93	93	235	236	241	247
100	90	89	234	235	240	246
Total loss in grams	113 5	115	71	69.3	65 8	68
Average loss in per cent..	56 0		23.0		21 5	

Rubber Covers. A cover in its lightest form is shown in Fig. 3-8 It is a sheath of compounded rubber about 1/40 to 1/32 inch thick; it resists abrasion to some degree, keeps out moisture, and helps to prolong the life of the friction rubber. When the material carried by the belt is heavy and the abrasion and cutting severe, the 1/32 inch is not enough; a heavier cover (Fig. 3-9) is needed to make a belt of balanced construction, that is, one that will get the benefit of the cover before the friction dries out and the plies tend to separate. On the other

hand, it is wasteful to use a cover too thick for the service; a belt whose useful life is limited by ply separation goes to the scrap pile with good unused rubber in its covers.

The purpose of a wearing cover is to protect the body of fabric *to an economical degree* from blows, from abrasion and cutting, and from the entrance of moisture. In giving this protection "to an economical degree" the cover must be neither too thin nor too thick. If too thin, the carcass may be bruised or cut so badly that the belt is thrown away while there is still good rubber in the cover. If the belt has a good friction and plies enough to transmit the pull easily and safely, and if the operating conditions are good, it pays to use a cover that will develop the full life of the friction rubber; but if the operating conditions are bad, if the belt is likely to suffer from neglect

FIG. 3-8.

FIG. 3-9.

FIG. 3-8. Eight-ply Elevator Belt, with 1/32-inch Rubber Cover on Both Sides.
FIG. 3-9. Eight-ply Elevator Belt with 1/8-inch Top Cover and 1/32-inch Cover on Pulley Side.

or accident, if the carcass is too light for the work, or if the quality of the friction is relatively low, then it is wasting money to put an expensive cover on the belt. The proper balance between the quality and thickness of the carcass and the quality and thickness of the cover differs according to the material handled, the care in loading and discharging, the hours of service, the length and speed of the conveyor, the importance of the installation, and the degree of care and maintenance it gets. These conditions are never the same in any two conveyors, but from the five standard grades of rubber belts offered by American manufacturers (Table 3-B), and also the belts sold in England to British Standard Specification (Table 3-C), it is almost always possible to choose a belt which, with the proper thickness of cover, will do the work in the best way.

Reinforced Covers. The adhesion between an ordinary cover and the frictioned ply of fabric next to it is generally less than the adhesion between two adjacent plies in the carcass because the rubber of the cover is not so plastic as the friction rubber and it does not get so good a hold on the fabric. In order to improve the hold of the cover to the carcass it is customary in the assembly of belts with high-grade covers to place under the top cover a layer or ply of strong open-mesh fabric previously frictioned and skim-coated on both sides. In pressing and vulcanizing, the rubber of the cover is pressed into and through the open-mesh "breaker ply." The breaker ply then acts in two ways; (1) to reinforce the rubber cover as, in a way, wire netting reinforces a slab of concrete pavement; (2) to prevent a gouge or tear in the cover from reaching the carcass, and thus prevent the cover from being torn loose from the carcass.

Tearing and Gouging of Rubber Covers. A strip of good cover stock ⅛ by ½ inch cannot be pulled to the breaking point by the strength of a man's hands, but by a tearing action starting at a nick

FIG. 3-10 Part of Cover Removed to Show Breaker Ply.

in one edge of the strip it can be pulled apart by the fingers. Covers can be torn when hard sharp pieces of material jam under skirt boards, when a chute fills up at the discharge end of a conveyor or in a hopper, when a conveyor is stopped while loaded and dribbles material into the gap between the belt and the end pulley, by chutes or tools falling down on the belt, or by carelessness in handling the belt in the roll or in pulling it into place over the idlers. Accidents like these will tear any belt, good or poor. If large pieces of the cover rip loose from the top ply with a clean separation, the belt may be defective, but it should not be condemned because the cover does not resist tearing or gouging. It is better that a cover should tear before it pulls loose from the carcass; a tear can be cemented or patched, but a loose cover can never be fastened on again to stay. A cover properly reinforced should not pull loose, but it is practicable to use the breaker-strip reinforcement only when the cover is $\frac{3}{32}$ inch or more in thickness. Makers' prices of high-grade belts generally include reinforcement of the top cover. Fig. 3-10 shows a piece of belt with some of the cover removed down to and exposing the breaker ply.

Cushioned Breaker Ply. For the hardest service where it is necessary to use covers $\frac{3}{16}$ or $\frac{1}{4}$ inch thick some manufacturers supply high-grade belts with the breaker ply imbedded in a layer of soft rubber compound known as "tie gum." This layer has little or no value to resist abrasion, but it has a very high friction value. It not only anchors the cover to the carcass very firmly, but what is even more important acts also as a resilient cushion which helps the cover throughout its life to dissipate the effect of severe blows and protect the fabric of the belt from bruises. Another point is that, if a cut should go through the top cover and through the tie gum, the softness and elasticity of the gum will tend to seal the cut and prevent moisture from getting into the fabric.

Samples of a 6-ply belt of this kind, tested in a standard machine (page 54), showed the following values of the "friction" between the layers of the belt:

Between	top cover and breaker ply	58 to 72 pounds
"	breaker ply and 1st ply of duck	20 to 32 "
"	1st and 2nd " " "	28 to 32 "
"	2nd " 3rd " " "	30 to 36 "
"	3rd " 4th " " "	25 to 29 "
"	4th " 5th " " "	25 to 30 "
"	5th " 6th " " "	27 to 33 "
"	6th " bottom cover	20 to 24 "

The belt was about 0.68 inch thick, top cover 0.19 inch, bottom cover 0.09 inch Note that the friction value of the tie gum is much higher than that of the friction between the plies.

For reasons stated above, it will be understood that thick covers are an important factor in increasing the life of a belt, but it is not so well understood that the increase in life is frequently more than proportional to the increase in the thickness of the cover. For instance, if a belt with a cover $\frac{1}{8}$-inch thick carried in its lifetime 400,000 tons of abrasive or cutting material, it may be estimated from experience that the outer $\frac{1}{32}$-inch thickness lasted for 220,000 tons, the next $\frac{1}{32}$ under it lasted for another 100,000 tons, the next $\frac{1}{32}$ for only 50,000 tons, and the last $\frac{1}{32}$ next to the carcass for only 25,000 tons. For important hard-worked belts, where the duck is under a high unit stress, it becomes very important to protect the carcass by a thick layer of good rubber. At one time a $\frac{5}{16}$-inch cover was considered very heavy, but now covers for important belts are made $\frac{7}{16}$ or $\frac{1}{2}$-inch thick. The protective effect of a thick cover is illustrated by the following. A certain wide belt with a $\frac{1}{4}$-inch cover, carrying abrasive ore, lasted for 6 million tons, a replacement belt with a $\frac{5}{16}$-inch cover

lasted for 9 million tons, a second replacement belt with a $\frac{7}{16}$-inch cover has carried 26 million tons and seems good for millions more. The covers of all these belts were fastened to the carcass by a breaker strip imbedded in the soft tie gum (page 43).

There is no fixed relation between the tensile strength of rubber and its resistance to tearing, although, in general, the higher the tensile strength, the better the resistance to tearing. This resistance is related in some way to the grain structure of the rubber and is a result of skill and knowledge in compounding and in manufacture. Good cover stocks resist aging very well; belt failures from that cause are not many.

Action of Wet. It is well known that a wet knife will cut rubber more easily than a dry knife. It is also true that hard sharp material in a wet condition will hurt a cover more than if the material were dry. When there is a choice, material should be handled dry, and as between keeping a belt dry by building a cover over it, and leaving it exposed to snow and rain, it is better to keep the belt dry.

Edges of Rubber Belts. If a rubber belt has its edge torn off or worn off, moisture and dirt get into the cotton duck, the friction rubber gives way, the plies separate, and the belt fails. Destruction of the edge may come from the use of side-guide idlers to correct bad alignment of the conveyor or to cure bad setting of the troughing idlers. The belt may rub against the chute at the discharge end, or it may interfere with idler brackets on the return run, or with the supporting framework of the conveyor.

There are several ways of making the edges of a rubber belt. Fig. 3-11 shows what was formerly used by several manufacturers. The outer ply was protected by a thickness of rubber continuous with the top cover and had an insertion of rubber under the outer ply to protect the assembled inner plies. The disadvantage was that the weave of the outer ply was not open enough to bond the rubber beyond it with the rubber enclosed by it, the edge was not firm enough, and the outer rubber could be torn off. A usual method of construction is to bring the top cover around the edge and join it to the rubber cover on the pulley side of the belt (Figs. 3-12 and 3-13). The 5-ply belt shown in Fig. 3-14 has a very thick edge to resist side wear, and a top cover $\frac{1}{8}$-inch thick held on by a layer of strong open-mesh fabric, "breaker strip," laid over the top ply, around the edge, and part way under the bottom ply. In belts of "plied" construction (Fig. 3-2), the rubber edge cushion is held to the carcass by being bonded with the friction rubber of all the plies. Sometimes some of the plies are made a little narrower than the others so as to roughen the edges

of the carcass and give a firmer hold for the edge rubber. It is customary to use breaker strip with the edge rubber in this form of belt.

Cord Construction. Cord construction for conveyor belts is shown in Fig. 3-15. It is recommended for long conveyors and hard-worked belts, particularly where heavy and hard lumps fall on the belt at the loading point. The fabric in standard belts is protected against injury from that cause by thick rubber covers, or reinforced rubber covers. In cord belts the few plies of fabric are made of very heavy duck to supply the necessary transverse strength and are placed near

FIG. 3-11.

FIG. 3-12.

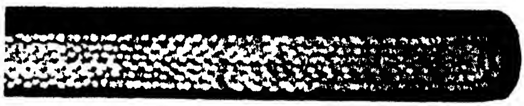


FIG. 3-13.

FIG. 3-14.

the pulley side of the belt where they are farthest removed from the shock of impact. Over these fabric plies are a number of layers of "weftless" cord fabric in which the warp cords are farther apart than the warp threads in belt duck. Since the cords are surrounded by rubber, the carcass of the cord belt acts better as a cushion than a standard belt does, yet it will stretch less because the cords are very strong and have been prestretched in the process of making. The cords are straight in the direction of belt pull, not crimped.

These belts can be run at unit stresses higher than belts made of standard duck. The makers state that metal splice plates are satis-

factory for ordinary service, but they recommend that the splices be vulcanized where it is necessary to develop the full strength of the belt. The Kimmich patent 1,735,686 of 1930 shows a vulcanized splice for a cord belt.

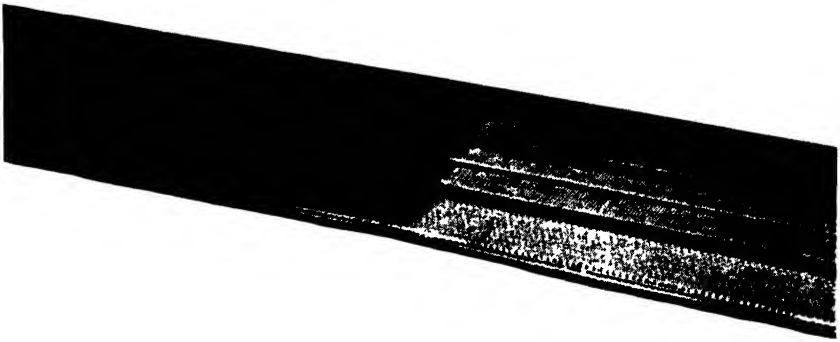


FIG. 3–15. Cord Construction Viewed Lengthwise of the Belt. (B. F. Goodrich Company.)

Stitched Edge Belts. When the alignment of a belt conveyor is bad and side-guide idlers are necessary, the wear on the edges of the belt is severe and the plies are likely to separate there while the body of the belt is still good. To prevent this separation, stitching the plies together along the edges may do some good. However, this has some drawbacks and the practice is not common because the stitching twine must be waxed for use in the sewing machine, and a rubber cover is more likely to loosen from it than from the threads of the duck.

Flanged Rubber Belts. In the Edison ore-concentrating plant (see page 11) from 1893 to 1896 there was serious trouble from the lengthwise cracking of the belts due to the steep angle (45°) of the troughing idlers. When the Edison cement plant was built a few years later, no belts were troughed; all were run flat, and some had light rubber flanges about ¾-inch high to retain the material. When these flanged belts ran through a tripper or over a reverse bend, there was a tendency to shear the edges off because the lower pulley of the tripper could not be made the full width of the belt, but had to clear the flanges of the belt. On the return run the flanges rubbed and chafed against the ends of the idler pulleys, which were narrower than the belt; as a consequence, the flanges were destroyed. Most of these belts were later replaced by troughed belts.

If the flanges had been made stiff and heavy, the edges of the belt might have been supported at the lower pulley of the tripper and on

the return run, but that construction is open to other objections and is costly.

Flanges (Fig. 3-16) are used on vanner belts employed in the wet concentration of metal-bearing ores. These belts are from 4 to 6 feet wide, made 2- or 3-ply with rubber flanges about 3/4 or 1 inch high; they run over end pulleys only 12 or 14 inches in diameter and about 10 feet centers. The speed is slow, usually less than 5 feet per minute, and the load is light, being merely a bed of water and ore about 1/4 or 1/2 inch deep. The stretch at the edge of a 3/4-inch flange in going halfway round a 12-inch pulley is 4 1/4 inches in 18 inches, and the rubber must be of very good quality to withstand it.

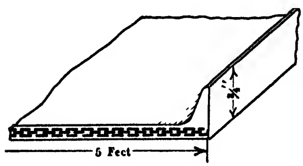


FIG. 3-16. Flanged Edge of Vanner Belt Used in Wet Concentration of Ores.

Flanged belts have also been used in ore-reduction plants to carry wet concentrates, the idea being that a troughed belt would not hold the wet and semi-fluid material so well and would be likely to spill. In some of these, as in the Edison plant, the support of the return belt gave trouble even when separate grooved idlers were placed at the ends of the return pulleys to support the flanged edges. Conveyors of this kind have no real advantage and they are seldom used now; a troughed belt of the proper width, and properly fed, works better than a flanged belt, and the cost for equal quality is considerably less, maintenance costs less, and since the parts are not special, repairs can be made with less delay.

In a vanner belt the carrying surface must be flat by nature of the process of concentrating the ore, and the flanges are necessary to retain the layer of ore and water; the flanges are always troublesome, however, and they should not be used on belts used solely to convey material.

Patents have been granted on a number of devices to increase the carrying capacity of a flat belt by turning up or flanging its edges, but none has come into commercial use. One of them (507,156 of 1893) covers a belt with side strips fastened on by flexible connections so that, while the strips normally lay in the same horizontal plane with the main belt, they could be turned up on the carrying run to form a trough-shaped section. Belts with sectional overlapping metal flanges have also been designed. Ridgway, in 1899, patented a belt with tubular edges which were intended to act as flanges to retain material on the belt and permit the belt to be run flat. If the tubes were bolted on or cemented on, they would pull loose; to vulcanize them to the belt

is difficult and costly. In no way is such a flanged belt better than that shown in Fig. 3-16, and where it has been tried, the results were no better than they were at the Edison plant in 1893.

In the manufacture of glue and gelatine the liquid material is poured on a moving flanged rubber belt; it jells or solidifies to a uniform thickness and then passes through heated ovens where hardening takes place. In canneries, flanged rubber belts carry fruits and juices without the aid of fixed side boards. There is no spill and less risk of contamination of food products carried on the belt.

A few makers in Germany furnish for portable conveyors and wagon loaders belts up to about 20 inches in width with rather high flaring flanges that make an angle of 120° with the belt. The flanges are made as separate pieces and are vulcanized on, not straight, but wavy or frilled so as to pass around small pulleys without much stretch on the outer edges of the flanges. The objects are to increase the carrying capacity of the belt on steep inclines and to reduce the weight of the machinery and of the portable loader itself.

Belts to Resist Heat. Ordinary rubber belts are likely to fail from ply-separation when they carry material at 150° F. or above, especially if the material is fine and holds its heat. If the pieces carried are large or if the load is a mixture of lumps and fines, the action on the belt is not so bad, because the heat is dissipated by air circulating through the material. There are rubber belts made especially for service in hot places or for carrying hot materials at temperatures not above 250° F. The rubber itself may be compounded especially for the work or it may not be vulcanized to the same degree as for ordinary service; or if there is danger that the carcass will actually be burned, the belt may have a cover with a layer of asbestos molded in it. When belts carry hot fines, a thick cover with an asbestos layer may not be worth its cost, but stitching the plies together will help to prolong the life of the belt. For such service a cover $\frac{1}{16}$-inch thick may be better and more economical than a thicker one. In handling hot materials on any kind of belt, rubber or canvas, the fabric must not get hotter than 300° F., because at that temperature cotton fiber begins to char and lose its strength.

Belts to Resist the Action of Oils and Chemicals. Rubber as usually compounded does not withstand repeated or continual exposure to oil. The rubber absorbs it, swells, and falls apart. Rubber can be compounded to show some resistance to such action, but Neoprene or some such synthetic rubber substitute will behave better; see page 50. Rubber can be compounded to resist the action of many of the ordinary acids and alkalies, but solutions of sulphuric acid in concentrations

above 50 per cent are destructive to any kind of rubber; so also are nitric acid and acetic acid even in weak form. At temperatures above 125° F. the deleterious action of such acids is greater.

Belts of Synthetic Rubber. Synthetic rubber, or, more properly speaking, synthetic hydrocarbon compounds which resemble rubber, have been made experimentally for many years. In Germany during the first World War some thousands of tons of "Buna" rubber were produced to meet the need caused by the scarcity of natural rubber. In this country since 1930, the E. I. du Pont Company has developed commercially a product called Neoprene, derived from acetylene. It resembles rubber in many of its qualities, can be compounded to show tensile strength comparable to that of rubber, is said to resist aging and the action of sunlight, and is not affected by oils which soften and ruin rubber. The recent practice of spraying coal with oil at the mines to keep down dust has caused some failures of rubber belts. In one case the absorption of oil by the cover of the belt caused it to soften and swell to such an extent that the normal concave of the belt over the troughing idlers became convex, and in order to carry the coal it was necessary, as an expedient, to turn the belt upside down. A belt with a Neoprene cover was put in which after 18 months' use was still unaffected by the oil. Present disadvantages of such rubber substitutes are: (1) the sources of supply are few; (2) the product costs more than rubber. The new belt mentioned above cost 80 per cent more than the old one. It is probable that the old one would originally have been supplied with an oil-resisting rubber cover had it been known in advance that the coal was oily.

Effect of Light, Heat, and Age. The absorption of oxygen by rubber causes what is known as drying or aging. It lessens the tensile strength and stretch of rubber and manifests itself by fine cracks in the surface of a belt. As the belt bends and stretches, the cracks widen and open, moisture gets into the fabric of the belt, and decay begins. Since the chemical change goes on faster in sunlight, it is advisable to store rubber belts in a dark place and to protect them from direct sunlight when in use. At a mine in Arizona a belt working in a conveyor gallery was considered worn out and was replaced by a spare belt of the same make which through carelessness had lain exposed to the sun for some months. The spare belt lasted only a few weeks; it was removed and the old worn belt was put on until a new belt could be obtained.

Belts age more rapidly when they are exposed to heat in storage or when they carry hot materials.

Figs. 3-17 and 3-18 show the effect of age on the tensile strength

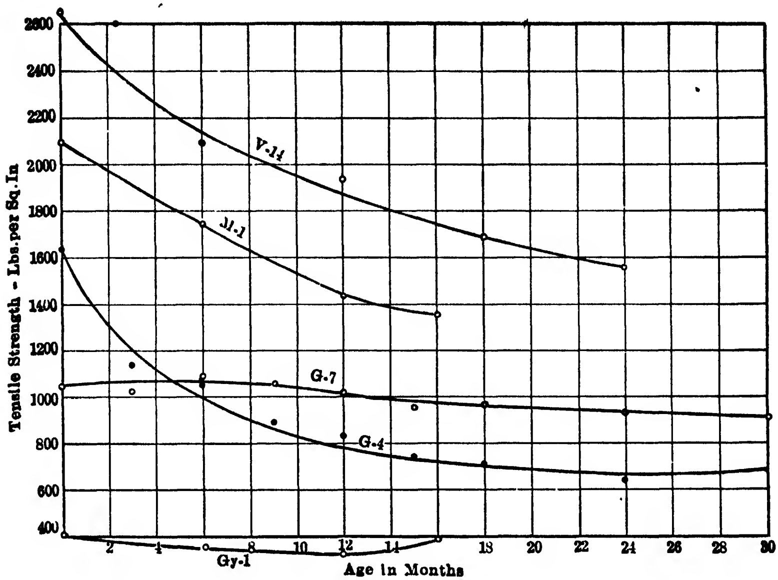


Fig. 3–17. Effect of Age on Tensile Strength of Five Specimens of Rubber Compounds. (Bureau of Standards, Washington, D. C.)

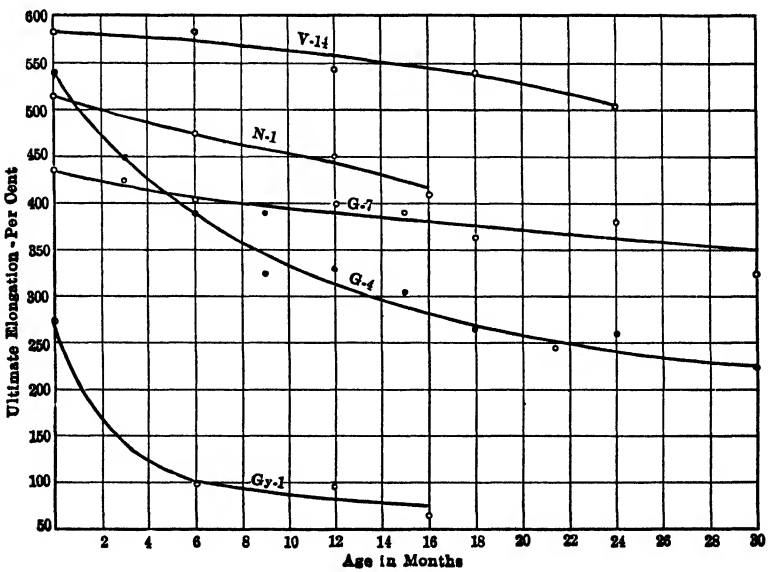


Fig. 3–18. Effect of Age on Stretch of Five Specimens of Rubber Compounds. (Bureau of Standards, Washington, D. C.)

and stretch of various rubbers, according to tests made by the Bureau of Standards. They do not refer to belts particularly, but merely show that deterioration of rubber is a natural phenomenon. It can be combatted to a certain extent by proper compounding and curing.

Much of the improvement in rubber-belt making in recent years has come as a result of a better knowledge of what to do to lessen the effects of heat, light, and age.

Specifications for Rubber Belts. Rubber belts came into use for handling grain about 1870; for years after that they were generally sold on the representations of their makers as to quality. The technique of belt manufacture was not well developed in the early years, but some of the belts were of remarkably good quality. The late Samuel W. Neall, who had charge of the Washington Avenue Grain Elevator in Philadelphia for many years, related the following: When the two galleries on the pier were erected in 1873, two 36-inch belts about 800 feet long were installed. In 1900 the two galleries were torn down and replaced by a single one twice as long; a new 36-inch belt was put in, and parallel to it a 1600-foot belt made by joining the two old ones. The new belt lasted seven years; the old belts were still there and in use when the elevator was dismantled in 1916, forty-three years after they were originally installed. When the Pennsylvania Railroad Company's Girard Point Elevator (Philadelphia) was torn down in 1916, the original 36-inch 4-ply rubber belts put in in 1882, thirty-four years before, were still in regular use. They were 700 feet long, and ran flat with portable concentrators at the loading point only.

As the business grew, the number of belt manufacturers increased and competition for business brought on the market many belts of poor quality. Probably some makers did not know how a good belt should be built. In the grain business this led to the use of detailed specifications for rubber belts. Typical of these were the specifications issued by the John S. Metcalf Company and by James Stewart and Company, Ltd., quoted in earlier editions of this book. A number of companies engaged in coal mining or ore mining brought out their own specifications, but as knowledge of belt conveying spread, and as belts improved in quality, the use of individual specification gradually waned, and now very few belts are bought in that way. On this point one manufacturer of rubber belting says:

> Skeleton specifications such as the duck weight, friction pull, and tensile strength of rubber covers are helpful to both buyers and manufacturers in a general classification of belt grades, but the buyer cannot depend on them as a measure of belt quality.

These specifications say nothing about quality of cotton, crimp of weave of the duck, the percentage of moisture in the duck at various stages of manufacture, elastic and inelastic stretch, resistance of the cover to tearing or abrasion, resistance of the rubber to aging, the details of belt assembly and fabrication. It is impossible to define these in specifications or to check them in the finished product. Methods of making belts are not standardized; each maker has developed his own. Some have progressed farther than others. Even if it were possible to write complete specifications and to devise tests to prove whether or not the specifications had been followed, purchasers would still be buying their belts on faith.

Strength of friction is in itself no sure index of the quality of the belt; if it were, the plies of fabric might be glued together and show a very high test, but such a belt would fail by cracking of the glue. In a similar way, a belt can be made with a rubber friction compound that will show a high friction test when fresh, but which will not keep its strength six months. For instance, a low-grade friction "doped" with rosin or shellac may show 18 pounds when fresh, but less than 10 pounds when ten months old. Then, too, the adhesion of the friction depends upon care in manufacture as well as upon the inherent strength of the rubber compound. The openness of the weave of duck, the degree of twist in the threads, the percentage of moisture in the duck when the rubber is pressed into it, the freshness of the frictioned surfaces when pressed together, the freedom of these surfaces from dust and the pulverized soapstone so freely used in rubber factories—all these factors are under control in the best establishments; but still all that the manufacturer can hope for is that his friction tests for a certain duck and a certain compound will fall within a range of 10 or 12 per cent above or below a desired standard.

FIG. 3–19. Test of Quality of Friction Rubber by Determining the Rate of Separation of Plies under a Given Weight.

Tests of Belt Friction. A simple way to make the friction test is shown in Fig. 3-19: lines 1 inch apart are marked on the strip; the plies are carefully separated for an inch or two and a 10-pound weight is hung to the lower end of the strip to keep it vertical. The pulling weight which is to measure the friction pull, whether it be 10, 13, or

18 pounds or more, is attached to one ply and the time rate at which it pulls loose is noted. The adhesion between the last two plies is not measured because, in spite of the 10-pound weight, the plies will bend at the point of separation and the pull will not be at 180° to the plane of the separating ply. The long spring is to steady the pull and prevent jerks.

Fig. 3-19A. Rubber Tester for Tensile Determinations. (Henry L. Scott Company.)

The time rate of separation is an important factor in these tests. It never varies directly as the pull. Up to a certain pull, there may be no separation at all, after which the rate increases gradually, then more rapidly, and finally a very small increase in the pull causes a great change in the rate of separation.

A machine used for testing the strength of friction in an accurate manner is shown in Fig. 3-19A. The pull required to separate the plies and the time rate of separation are automatically recorded on the chart. The machine can also be used to test rubber, yarns, and various other articles. For references to standard methods of testing, see page 56.

Standard Rubber Belts. The Technical Committee of the Mechanical Rubber Goods Manufacturers Division of the Rubber Manufacturers Association, Inc. (United States), has brought about a degree of standardization of belts based upon the adoption of four grades of covers and three grades of friction. American manufacturers still sell their belts under trade names of their own, but as to quality they fall into fairly definite classes, as shown in Table 3-B.

British belt manufacturers have adopted the recommendations of the British Standards Institution specification No. 490, 1933, for Rubber Conveyor and Elevator Belting. This gives three grades of cover and three grades of friction; see Table 3-C.

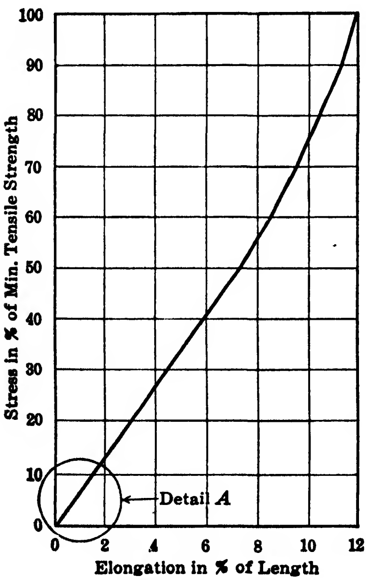
TABLE 3-B

American Standard Grades of Rubber Belts

Grade	Rubber Cover		Duck Nominal Weight, ounces	Friction	
	Tensile Strength of Cover Stock, pounds per square inch	Minimum Elongation, per cent		Minimum Adhesion between the Plies, pounds per inch width	Minimum Adhesion Cover to Carcass, pounds per inch width
1	3500–4000	500	28, 32, 36, 42	20	17
2	2500–3000	450	28, 32, 36, 42	16	13
3	1400–2000	350	36, 42	16	13
4	1400–2000	350	28, 32	12	10
5	800–1000	250	28, 32	12	10

Relation between Stress and Elongation. Fig. 3-20 shows the results of tensile tests of standard American rubber belts, and also

Per Cent Minimum Tensile Strength	10	20	30	40	50	60	70	80	90	100
Per Cent Elongation	1.5	3.0	4.5	5.9	7.3	8.5	9.5	10.4	11.3	12.0



Nominal Weight of Duck in ounces	Minimum Tensile Strength per inch of width per ply in pounds
28	280
32	300
36	325
42	375

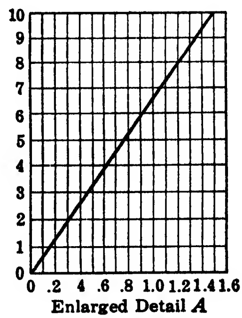


Fig. 3-20. Relation between Stress and Elongation for Rubber Conveyor Belting. (Rubber Manufacturers Association, October, 1937.)

the relation between stress and elongation. The average stretch at the breaking point is 12 per cent, but within the usual working limits in which rubber belts are used in service, that is, at not above 15 per cent of the breaking stress, the elongation is about 0.15 per cent for

TABLE 3-C

RECOMMENDED BRITISH STANDARDS

Grade	Rubber Cover		Friction	
	Tensile Strength, pounds per square inch	Elongation, per cent	Adhesion, Ply to Ply, strip 1 inch wide, separation 1 inch per minute, pounds	Adhesion Cover to Ply, strip 1 inch wide, separation 1 inch per minute, pounds
A	3500	550	26	24
B	2500	450	20	18
C	1500	350	17	15

each 1 per cent of the ultimate strength of the belt. Table 3-D derived from Fig. 3-20 gives the tension in pounds per inch per ply for various weights of belt ducks at different percentages of their ultimate strengths.

Tests of Rubber Belting. The Technical Committee of the Mechanical Rubber Goods Division, see above, states that the following Specifications of the American Society for Testing Materials (A.S.T.M.) are used by American manufacturers in their own establishments to control the quality of the raw materials:

D181-36 Standard Methods of Testing Hose Ducks and Belt Ducks.
D39-36 Standard General Methods of Testing Woven Textile Fabrics.
D76 Standard Specifications for Textile Testing Machines.
D15-35T Tentative Methods of Physical Testing of Rubber Products.
D412-36T Tentative Methods of Tension Testing of Vulcanized Rubber.
D413-36T Tentative Methods of Testing for Adhesion of Vulcanized Rubber. (Friction Test.)

The Committee says that tests of raw duck or of frictioned duck are not a true index to the strength of finished belt of such duck. It recommends that tests of the finished belt be made as follows:

> For belting up to 5 inches width inclusive, a longitudinal test specimen 20 inches long and the full width shall be cut off. For belting

TABLE 3-D

STRETCH OF RUBBER BELTS AT VARIOUS TENSIONS

Tension in Belt as Percentage of its Strength Per Cent	Stretch of Belt as Percentage of its Length Per Cent	Tension in Pounds per Inch per Ply			
		28-ounce Duck	32-ounce Duck	36-ounce Duck	42-ounce Duck
		Standard Strength 280 lb.	Standard Strength 300 lb.	Standard Strength 325 lb.	Standard Strength 375 lb.
1	0 15	2.8	3 0	3.25	3.75
2	0 30	5 6	6 0	6.50	7 50
3	0 45	8 4	9.0	9 75	11 2
4	0.60	11 2	12.0	13 0	15 0
5	0.75	14.0	15 0	16.2	18 7
6	0 90	16 8	18 0	19.5	22 5
7	1 05	19 6	21 0	22.7	26 2
8	1 20	22 4	24 0	26 0	30 0
9	1 35	25 2	27 0	29 2	33 7
10	1 50	28 0	30 0	32 5	37 5
11	1 65	30 8	33 0	35 7	41 2
12	1 80	33 6	36 0	39 0	45 0
13	1 95	36 4	39 0	42 2	48 7
14	2 10	39 2	42 0	45 5	52 5
15	2 25	42 0	45 0	48.7	56 0
16	2 40	44 8	48 0	52.0	60 0
17	2 55	47 4	51 0	55.2	63 7
18	2 70	..	54 0	58 5	67 5
19	2 85	. ..	.	61 7	71 2
20	3 00	. ..	. . .	.	75.0

The heavy upper lines indicate unit stresses in pounds per inch per ply up to which it is considered safe to work belts steadily. See also Table 6–B, page 169.

The light lower lines indicate unit stresses which may be permitted for occasional overloads.

over 5 inches wide, the specimen shall be 20 inches long by 5 inches wide, and the edges of the specimen shall be cut parallel to and at least ½ inch from the edges of the belt. The thickness of the test specimens shall not exceed 4 plies. Excess plies shall be removed by stripping, removing them alternately from each face. When testing rubber-covered belts, rubber covers $\frac{1}{16}$ inch and over in thickness shall be stripped from the test specimen. . . . Specimens shall be tested in a suitable vertical testing machine equipped with wedge-shaped jaws. The rate of separation of the jaws shall be

from $\frac{3}{4}$ inch to 2 inches per minute inclusive. . . . The unit of tensile strength determined by this method shall be expressed in pounds per inch of width per ply of duck. The nominal weight of duck is based on a length of 36 inches and a width of 42 inches. All regular and special ducks are classified soft or hard and then subdivided in each group by weights as indicated below:

Nominal Weight of Duck	Minimum Tensile Strength per Inch per Ply, pounds
28-ounce soft duck	280
32-ounce soft duck	300
36-ounce soft duck	325
33-ounce hard duck	300
35-ounce hard duck	320
42-ounce soft duck	375

For permissible working tensions of rubber belts, see Chapter 6.

The British Standard Specification for Rubber Conveyor and Elevator Belting No. 490, 1933, mentioned on page 54, can be obtained from the British Standards Institution, Publications Department, 28, Victoria Street, London, S.W.1, London, England, price 2 shillings 2 pence; or through the American Standards Association, 29 West 39 Street, New York City, for 75 cents.

The following publications by the United States Government can be obtained from the Superintendent of Documents, United States Government Printing Office, Washington, D. C., at the prices stated:

Federal Specification ZZ-B-206 Belting; Conveyor (Rubber). This refers particularly to belts used by the U. S. Post Office Department. Price 5 cents.

Federal Specification ZZ-R-601 Rubber Goods: General Specifications. (Methods of physical tests and chemical analyses.) This refers to rubber goods of all kinds. Price 5 cents.

Circular of the Bureau of Standards No. 38. This gives useful information about the manufacture of rubber goods, methods of testing, use of testing machines, etc. Price 30 cents.

Specifications issued by the American Society for Testing Materials can be obtained from the office of the Society at 260 South Broad Street, Philadelphia, Pa.

How to Keep Rubber Goods. The following information is furnished by the B. F. Goodrich Rubber Co:

No matter how good the quality of rubber, or how short a time it is to be kept for use, it is worth while to give some attention to

storage conditions. The oxidation of rubber is promoted by the presence of heat, light, and air. Ordinarily, room temperatures are satisfactory except close to the ceiling or in a room under the roof. Basement rooms are preferable, and the ideal storage place is a cool, dark cellar protected from frost in winter and heat in summer, yet not too close to the heating apparatus. Dry air deteriorates rubber much more rapidly than moist air, though an excess of moisture is not best for articles like belts which are made partly of fabric. A cellar storeroom is better if the floor or walls are natural earth, provided that there is sufficient ventilation to avoid excessive moisture.

Since the air in artificially heated buildings in winter is usually excessively dry, it is advisable, where practicable, to provide some special means to maintain normal humidity in the storeroom. Some users of rubber goods provide humidors in the form of closed bins or boxes in which are kept bricks previously soaked in water, or flat trays kept filled with water.

In cases where it is not convenient to provide special storage arrangements, care should at least be exercised to avoid placing rubber belts and other rubber goods near steam pipes, radiators, hot-air registers, windows, or ceilings.

Occasional Defects in Rubber Belts.

1. Longitudinal seams in the wrong place, duck not cut to right width, not folded properly.
2. Open seams or plies not meeting at the butted edges, requiring a filler strip to close the gap.
3. Blisters between the plies or under the cover due to moisture in the duck at time of vulcanization.
4. Rubber covers or edges damaged by careless handling.
5. Soft edges, outside ply not backed up by inner plies.
6. Outside plies broken by overstretching.

Weight of Rubber Belting. Weights are given in tables in manufacturers' catalogs. They are based on the following:

TABLE 3–E

Duck, ounces	Carcass Weight per Inch Wide per Ply of Thickness, pounds	Cover Weight per Inch Wide, per $\frac{1}{32}$ Inch Thick, pounds
28	0 021	0 018
32	0.024	0.018
35, 36	0.026	0.018
42	0.029	0.018

Example. What is the weight per running foot of a 36-inch-wide 6-ply belt of 32-ounce duck with ⅛-inch top cover and 1/16-inch pulley cover?

Carcass	$36 \times 6 \times 0.024$	= 5.18 lb.
Top cover	$36 \times 4 \times 0.018$	= 2.59 lb.
Pulley cover	$36 \times 2 \times 0.018$	= 1.29 lb.
Weight per foot of belt		= 9.06 lb.

Thickness of Rubber Belts.

TABLE 3-F

	28-Ounce Duck	32-Ounce Duck	36-Ounce Duck	42-Ounce Duck
Thickness of one ply, in inches	0 045	0 053	0 056	0.063

Example. What is the thickness of the belt mentioned in the example above?

Thickness of carcass	6×0.053	= 0.32 in.
Thickness of covers	⅛ plus 1/16	= 0.19 in.
Thickness of belt		= 0.51 in.

Diameter of a Roll of Belting. The area of the circle formed by the rolled-up belt is the area of the edge of the belt plus the area of the hole in the middle of the roll. The latter is usually from 20 to 30 square inches.

Example. What is the diameter of a roll of 500 feet of belt mentioned above?

Area of edge equals $500 \times 12 \times 0.51$		= 3060 sq. in.
Area of hole,	say	= 30 sq. in.
Total area		= 3090 sq. in.
Diameter		= 62¾ in.

For the diameter of the crated belt, add 4 or 5 inches.

STITCHED CANVAS BELTS

Stitched canvas belts used for conveyors and elevators are made of plies of cotton duck stitched together in a sewing machine and then made waterproof. Most of them are of folded construction; some are of "plied" construction. For the latter the duck is woven with a selvage to the width of the belt. The duck for light service weighs 33.6 ounces for a piece 42 inches wide by 36 inches long, but 37½-ounce

duck is used for most canvas belts. The corresponding weights for a piece 1 yard square are 28 and 32 ounces, respectively.

There are various weaves of 37½-ounce duck, depending on the service expected of the belt and on the methods of the belt maker. In general the warp threads are closer, say 28 per inch, than the filler threads, which may be 16 per inch. Usually the threads are all of the same size, but some belt makers prefer to have the filler threads of conveyor belts heavier than the warp threads. As a rule, duck for canvas belts is more closely woven and harder than duck for rubber belts.

Fig. 3-21 is a diagram showing, in a conventional way, the construction of a 6-ply stitched belt for elevator or conveyor service.

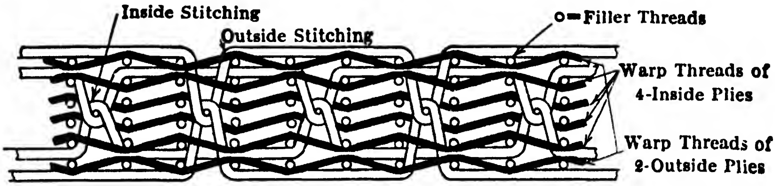


FIG. 3-21. Diagram Showing Assembly of a 6-Ply Stitched Canvas Belt.

Duck of the proper width is folded and assembled to make the 4 inside plies, and these are stitched together by rows about 1 inch apart. Then the wrapper or cover is added to make 2 more plies and the whole belt is sewed again; this time the stitches are in rows ¼ inch apart. The thread is a strong cotton twine, heavier than the warp or filler; the lockstitch between the needle thread and the bobbin thread of the sewing machine is buried within the thickness of the belt and it is not likely to come loose even when the threads are worn away on either side of the belt. In that respect the sewing is like that which fastens on the sole of a shoe; the sole is held on even after the exposed stitches have been worn away.

The next process with some makers is to subject the assembled plies to a combination of pressing and stretching, after which the belt is immersed in the waterproofing compound. It is then run between rolls to squeeze out the excess of liquid. Some makers stretch the belt during this operation, but in all factories the wet belts are finally dried and "cured" by stretching them out in horizontal spans 150 feet or more in length and keeping them under tension for some days or weeks to season. During this time the oil in the compound dries or partially oxidizes to a gum; when linseed oil is used, the gum formed has less strength and less elasticity than rubber, but it is at the same time less affected by light, heat, and age. To a certain extent, it acts like the

oxidized linseed oil which forms the basis of floor linoleum, and when properly applied and "cured" it gives a belt much greater resistance to abrasion than the raw cotton possesses. Belts saturated with mineral oils, semi-drying oils, and other substitutes for linseed oil are more flexible, but they do not resist wear so well and are more likely to stretch in service. Belts that are not maintained under stretch in the factory long enough to dry or set the oil properly will also show excessive stretch and a shorter life. The thicker the belt, the longer it should be cured; a good 8-ply belt may be kept under stretch for several months before it is considered ready to use. The output of a belt factory can be increased and the cost of the product decreased by shortening the time of "cure," but it is likely to be at the expense of quality in the belt.

Saturating compounds for canvas belts are of various kinds depending on the work of the belt. (Information from Imperial Belting Company, Chicago.)

Class 1. Drying Compounds. Belts for ordinary elevator and conveyor service are generally soaked in a thin oil compound, the basis of which is linseed oil or some other drying or oxidizing oil. It is thin enough to penetrate all the fibers of cotton in the belt and after the excess has been squeezed out, a definite quantity of it remains in the fibers to waterproof them to a certain degree, to lessen the wear due to internal bending and friction, and when properly "cured" to make the threads of warp and filler tougher and more resistant to abrasion. Such belts are then coated with a paint made of drying oil and a mineral pigment; this fills up the stitching holes and, when dry, gives the stitching twine and the outside plies of the belt a tough, hard surface. This surface coat also gives the belt a good coefficient of friction for contact with the pulley, and it prevents the saturating liquid remaining in the cotton fibers from drying out during the life of the belt. Belts treated with Class 1 compounds are not suitable for working in the wet, or under great heat, or where they are subject to acid or alkali dust or fumes.

Most canvas belts in heavy conveyor and elevator service are treated with compounds of this class. They are used to carry coal, stone, ore, sand, gravel, clay, particularly if the materials are dry. They are not harmed by moderate exposure to the weather or by contact with oil-treated coal or with the mineral oils used as lubricants and coolants in shops. They do not withstand temperatures steadily over 150° F.

Class 2. What are known as asphalt compounds are mixtures of asphalt or gilsonite with various oils and gums. They make a belt

that withstands water, heat, and chemical action, but does not resist surface wear so well as those treated with Class 1 compounds. These compounds are not likely to dry out or harden with age.

Class 3. Colorless waterproofing compounds are used for belts that convey bread, crackers, and other food products, wrapped packages in stores, books, fabrics, etc., which must not be discolored in handling. Compounds of Class 1 and Class 2 are open to objection in this respect. Class 3 compounds leave the canvas its natural color and give sufficient protection against atmospheric moisture; they do not give a belt great resistance to the rough handling of heavy packages or to the actual contact of wet material. They make a belt that will run over small pulleys and remain flexible throughout its life.

Class 4. Tasteless, odorless, colorless compounds similar to Class 3 are put into belts for handling bread, sugar, candy, cereals, and similar foodstuffs. They contain a wax which protects the cotton against the action of fruit acids, and being light in color, they give the belt that appearance of cleanliness which is desired in canneries. The compound is not affected by hot water or steam which is used in cleaning such belts to keep them sanitary. A coating of cellulose solution is sometimes brushed on one or both sides of such belts to make them easy to clean.

From what is said above it is apparent that the duck in nearly all canvas belts is of the one kind, but that the saturating compounds are several, and can be chosen to give the belt the desired qualities and fit it for its particular work. There is also some choice in making the surface of the belt; some belts have a high coefficient of friction and get a good grip on the driving pulleys; others have a low coefficient of friction to permit bags, boxes, etc., to be deflected sideways from the belt, or to allow the belt to slide easily on runways, or to avoid damaging packages if they pile up in the discharge chute at the end of the conveyor and foul the belt.

Stitched Canvas Belts for Conveyors. Canvas belts for handling bulk materials and heavy packages are generally treated with a Class 1 compound and when properly made they stretch no more than a rubber belt and possess considerable resistance to cutting and abrasion. This resistance is less than that of a belt with a rubber cover, but the canvas belt is homogeneous whereas the rubber belt is not. The cover and the friction rubber in a rubber belt protect the raw cotton duck from wear, and when the wear has gone through the cover and into the duck, the plies are held together only by the tenacity of the friction gum. This rubber deteriorates on exposure to the air, and the raw cotton is liable to decay from the action of moisture and dirt. In a properly

treated fabric belt, the whole thickness is equally resistant to abrasion, the plies are held together by stitching, which is to a certain extent effective even when the outer plies are worn away, and the threads and fibers of the duck do not mildew from the absorption of moisture, as raw cotton does. When the cover of a rubber belt and its first friction layer are worn away, it approaches the end of its life, but a canvas belt is intended to expose its whole thickness to wear, and it should bear the loss of several of its plies before it is discarded. For a discussion of the conditions under which stitched canvas belts give good service at relatively low cost, see page 70.

For carrying boxes, packages, mail bags, and parcel-post goods in mail-order houses and in United States post offices stitched canvas belts may be said to be "standard." The requirements of the United States Post Office Department are stated in Federal Specifications DDD-B-166 and DDD-B-171 sold by the Superintendent of Documents, United States Government Printing Office, Washington, D. C.; price 5 cents each.

Canvas Belts for Hot Materials. A stitched belt impregnated with the proper compound will resist to a good degree the action of hot bulk material such as foundry sand, hot cement, or hot lime up to temperatures of 300° F. For handling material containing very hot or red-hot pieces which would burn the fabric it is possible to get belts with a top ply of asbestos cloth to protect the inner plies.

Canvas belts are not made with rubber wearing covers, but can be made with felt covers sewed or cemented on for carrying crockery, glassware, and other fragile articles. Belts with a coating of latex applied in liquid or paste form have a high coefficient of friction and will carry packages or boxes on angles steeper than 20°, even up to 35° or 40°.

For weights of stitched canvas belts see Table 3-G.

Solid-woven Cotton Belts. Unlike other belts, these are not woven like duck, a ply at a time, but are woven in their full thickness in looms built for the purpose. Layers of warp (lengthwise) threads under tension are woven in with layers of filler (crosswise) threads to make the required thickness and the whole mass is bound together in the loom by lines of binder threads which pass around the filler threads from face to back of the belt. Fig. 3-22 shows, in a diagrammatic way, sections lengthwise and crosswise of such a belt. The binder threads serve the same purpose as the stitching used in a canvas belt (see Fig. 3-21); without them the belt will not hold together.

Solid-woven belts are not sold on specifications as to size of threads, weight of assembled belt, or kind of weave. They are known by

various trade names, and the thicknesses are designated by terms which are somewhat arbitrary. Belts for conveying and elevating

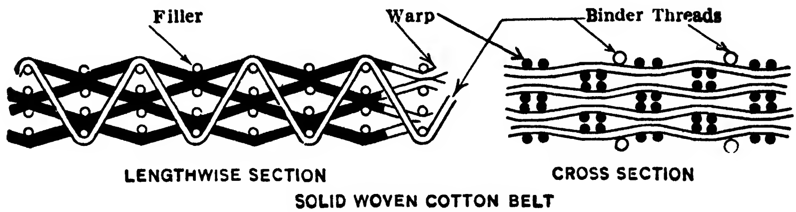


FIG. 3–22. Diagram Showing Assembly of a Solid-woven Belt.

service have generally 6 or 8 layers of warp threads and for tensile strength correspond to 6- or 8-ply canvas or rubber belt.

TABLE 3–G

WEIGHT OF OILED AND PAINTED STITCHED CANVAS BELTS MADE OF $37\frac{1}{2}$-OUNCE DUCK

(Main Belting Company)

For asphalt-treated belts add 10 per cent.

Width, inches	Weight of 1 Foot of Belt, Pounds						
	4-ply	5-ply	6-ply	7-ply	8-ply	10-ply	12-ply
12	1.30	1 63	1.95				
14	1.52	1 90	2.28				
16	1.73	2.17	2 60	3 04			
18	1.95	2.44	2.93	3.42			
20	2 17	2.71	3.25	3.79			
22	2.39	2.98	3.58	4.17	4.77		
24	2.60	3 25	3.90	4.55	5.20		
26	2.82	3 52	4.23	4 93	5.64	7.05	
28	3 04	3.79	4.55	5 31	6.07	7.59	
30	3.25	4 07	4.88	5.69	6.50	8.13	
32	3.47	4 34	5 20	6.07	6.94	8.67	
34	3.69	4 61	5 53	6.45	7.37	9.21	
36	3.90	4 88	5.85	6 83	7.81	9.76	11.71
38	4.12	5 15	6 18	7 21	8.24	10.30	12 36
40	4.34	5.42	6 50	7 59	8 67	10.84	13 01
42	....	5 69	6 83	7 79	9 11	11.38	13.66
44	....	5.96	7 15	8 35	9 54	11.92	14 31
46	. .	6 23	7 48	8 73	9 97	12 47	14.96
48	....	6.50	7.81	9 11	10.41	13.01	15.61

Tables 3-H and 3-J give facts about two makes of solid-woven belt.

TABLE 3-H

"WOOSTER" SOLID-WOVEN COTTON BELTING

(Duryea Products Company, Jersey City, N. J.)

Kind of Belt	Average Strength, pounds per inch width	Average Weight per 1 Inch Wide per 100 feet, pounds	Average Thickness, inch	Equivalent in Stitched Canvas or Rubber Belt
Light......	800	11	$\frac{3}{16}$ to $\frac{7}{32}$	4-ply
Medium.....	1400	14	$\frac{1}{4}$ to $\frac{9}{32}$	6-ply
Heavy......	2300	17	$\frac{5}{16}$ to $\frac{3}{8}$	8-ply

TABLE 3-J

"SCANDINAVIA" SOLID-WOVEN COTTON BELTING

(Scandinavia Belting Company, Newark, N. J.)

Kind of Belt	Average Strength, pounds per inch width	Average Weight per 1 Inch Wide per 100 feet, pounds	Average Thickness, inch	Equivalent in Stitched Canvas or Rubber Belt
Single...	1200	8 5	$\frac{3}{16}$	4-ply
Double, up to 8-in ..	2200	13	$\frac{1}{4}$	5-ply
Double, above 8-in	2400	15	$\frac{5}{16}$	6- or 7-ply
Triple.... .	3200	18.5	$\frac{3}{8}$	8- or 10-ply

Solid-woven belts were made originally for power transmission, and for that purpose they are most often used. When impregnated they are treated with a Class 2 waterproofing compound and are cured by stretching them out in long spans in the way stitched canvas belts are treated. As they come from the loom they are more flexible than stitched canvas belts, and with a Class 2 impregnation, they retain that characteristic and hence conform very well to the contour of standard troughing idlers. They do, however, lack the hard surface of painted canvas belts and are not so tough as those treated with Class 1 compound; and in many of them the binder threads are lighter than the stitching threads used in canvas belts. They are likely to

stretch more in service than rubber belts and canvas belts, but they have a high coefficient of friction for contact with the driving pulleys.

TABLE 3-K

TESTS OF SEVERAL BELTS OF KINDS USED FOR PACKAGE CONVEYORS

Total Load, pounds	Per Cent Stretch in $6\frac{1}{4}$ In.		
	Solid Woven 6-in. × 4-ply, Untreated		Stitched Canvas 6-in. × 4-ply Class 1 Impregnation
	No. 1	No. 2	
600	8.4	7 3	1.9
1200	11 3	10.1	2.7
1800	12.6	12.3	3.4
2400	14.3	13.6	4.3
3000	15.6	14.8	5.1
3600	17 0	16 4	5.8
4200	18.2	17 3	6.9
4800	18 7	18.2	7.6
5400	Broke at 5330 lb.	Broke at 5230 lb.	8.9
6000	......	....	9.5
6600	...	.	10.8
7200			Broke
Breaking strength in pounds per inch width per ply .	222	218	300

These tests were made in a 50,000-pound Riehle machine. The movement of the jaws was not stopped from the time the machine started until the belt broke The record of stretch was taken electrically at the increments of load indicated in the left column Belt specimens were 18 inches long, 10 inches clear between jaws of machine.

Solid woven belts are frequently used without waterproofing for light package-conveyors in stores where they are always under cover and not subjected to changes of temperature or severe pulls. Table 3-K compares two specimens of such belting and one stitched canvas belt in respect to stretch and ultimate strength.

Solid-woven Pulley Lagging as made by the Victor Balata and Textile Belting Company, Easton, Pa., is an asphalt-impregnated belt with a rough face, like the surface of a Turkish towel. A piece of it cemented and bolted to the rim of a pulley gives a good contact with the conveyor or elevator belt and makes it less likely to slip when it is wet or coated with frost.

"R. F. & C." (rubber filled and covered) belts are made by the Buffalo Weaving and Belting Company, Buffalo, N. Y. They are solid-woven cotton belts impregnated with a rubber solution and enclosed in a rubber cover. For conveyor work the top cover is made thicker than the cover on the pulley side of the belt. By laboratory test they are more nearly waterproof than other kinds of belt.

Latex-treated Belts. Some manufacturers advertise solid-woven belts coated or impregnated with a rubber latex compound. These are used in such places as canneries, where a belt should be waterproof and steam-proof, and should not contaminate articles of food.

Balata belts are made of cotton duck waterproofed and held together by balata, a tree gum brought from the West Indies and the north coast of South America. The gum is plastic at 212° F., stronger than rubber at ordinary temperatures, but not so elastic. It does not absorb oxygen from the air to the extent that rubber does; hence it retains its life and strength for a long time.

The raw gum is washed, as rubber is, to remove dirt, then dried, cut up, and dissolved in a liquid solvent which, when applied to the duck, carries the gum into the threads and fibers of the cotton. The duck is generally of a closer weave than that used for rubber belts; in some balata belts it weighs 28 to 32 ounces per square yard, but as made for conveyor and elevator service the duck in the best balata belts weighs 38 to 40 ounces and has the warp (lengthwise) threads somewhat heavier than the filler (crosswise) threads.

In making the belt, the duck is washed to get rid of any oil or sizing used in spinning or weaving the yarn; then it is dried, run through the solution of gum, and dried again. If the belt is of folded construction, the duck is cut to the right width, folded, refolded, and rolled under pressure while warm to build up the required thickness. Some balata belts are of "plied" construction in which each ply of duck has been woven with a selvage to the width of the belt. With either construction the belt is then stretched and allowed to cool under strain. This makes a dense, strong belt, rather stiff, but not likely to stretch, and since its fibers are impregnated with the gum, it shows a high resistance to the absorption of water (see Table 3-M). The gum softens at 120° F., and the belt must not be used where it may get too hot.

Balata belts have been made in Great Britain with rubber covers fastened on by cement and by rows of copper-wire stitching, but that style is not known in this country. Balata belts have been used for conveyor work to a greater extent in Europe than in this country; this may be attributed chiefly to the fact that the manufacture of rubber

belts has received more attention in this country than anywhere else and competition from other kinds of belt has been made more difficult here than abroad. In this country balata belts compete with others more actively as transmission belts and elevator belts, in the latter service because of their great strength and freedom from stretch, and especially in elevating wet materials like mineral pulps where the waterproof quality of the balata belt is an advantage. In conveyor service they do not trough so well as rubber belts because of their density and stiffness; hence they are better suited to run on shallow troughing idlers or flat rollers rather than on three-pulley or five-pulley idlers. Table 3-L shows weights of balata belts.

TABLE 3-L

WEIGHT OF BALATA BELT (38-OUNCE DUCK)

(R. & J. Dick Company)

Width, inches	Weight of 1 Foot of Belt, pounds						
	3-ply	4-ply	5-ply	6-ply	7-ply	8-ply	9-ply
12	0.83	1 11	1 40				
14	0 97	1 30	1 63				
16	1.11	1 48	1 85	2 22			
18	1 25	1 67	2 09	2 52			
20	1 39	1 85	2 31	2 79	3 24		
22	1 53	2 04	2 54	3 06	3 56		
24	1 67	2 22	2 76	3 33	3 88	4 43	
26	1 80	2 40	3 00	3 61	4 20	4 80	5 40
28	1 94	2 59	3 23	3 88	4 52	5 17	5.81
30	2 08	2 77	3 46	4 16	4 84	5 54	6.23
32	2 22	2 96	3.69	4 43	5 17	5 91	6 64
34	2.36	3.14	3 92	4 71	5 49	6.28	7.06
36	2 50	3 33	4 15	4 99	5 81	6 65	7 47
38	2 64	3 51	4 38	5 26	6.14	7 02	7.89
40	2.78	3 70	4 61	5 54	6 47	7.39	8.31
42	2.91	3 88	4 85	5 82	6 79	7.76	8.73

Absorption of Water by Various Belts. Table 3-M gives results of laboratory tests of the absorptive capacity of various kinds of belts. The specimens were all 4-inch 4-ply belts cut 5 inches long and treated alike. The figures in the table may not show which kind of belt absorbs the least moisture in actual service when subjected to stretching and bending in the presence of water; under those conditions and after

a period of service, the belt becomes more pliable, the body becomes more open, and the percentage of absorption will be greater with belts of all kinds. Nevertheless the greatest absorption in service will be shown by those belts which give a high percentage in the laboratory tests, and to that extent the figures of the table are a useful guide.

TABLE 3-M

ABSORPTION OF WATER BY VARIOUS KINDS OF BELT

Four-inch, 4-ply belts, 5 inches long, soaked 50 hours in water at 75° F.

	Per Cent of Weight of Belt
Various rubber belts	4 to 10
Standard grade stitched and painted canvas belts	9 to 11
Cheaper grade stitched and painted canvas belts.	12 to 22
Belts saturated with asphalt compounds	3 to 4
Untreated cotton belts	30 to 40
Balata belts .	5 to 9
Rubber-filled and covered belt	2.7 (1 specimen)

Various Belts for Different Kinds of Service. American practice in belt conveying has for years been close to the rubber-belt business, and more rubber belts are used for conveyor work than belts of other kinds—stitched canvas, balata, or solid-woven cotton belts. Rubber belts can be made to handle economically material of all kinds, light or heavy, fine or coarse, wet or dry. For carrying hard material in heavy pieces a high-grade rubber belt with a good cover should make the best conveying medium, but it would not be economy to use such an expensive belt for carrying crushed coal, and it would be a waste of money to use it on a package conveyor. A rubber belt of medium grade with a light cover will usually carry small coal for less cost per ton or per year than a high-grade belt, especially since the life of a belt so often depends upon factors external to the belt itself.

In conveyors where there is no severe cutting action from the impact of material at the loading point, and where the belt is protected from the weather, it is often possible to use canvas belts with economy. The relative prices of canvas belts and rubber belts depend largely on the costs of raw cotton and raw rubber. Since these fluctuate, the ratios of prices are not constant, but, in general, canvas belts cost less than rubber belts of equal width and ply. No one thinks that a canvas belt resists cutting and abrasion so well as a rubber belt with a cover, but when its lower cost is considered, it may be economy to use it and get less tonnage or a shorter life than to pay much more for a rubber belt. On the other hand, buying canvas belts for some conveyors

would be throwing money away; the material may be too heavy or too sharp, the conditions of loading and discharge, lubrication, care, and general oversight may be so good that a rubber belt will have a chance to show a life much greater in proportion to its cost than any other kind of belt. In other words, if a canvas belt costs $750 and under the operating conditions lasts one year, it is better to pay $1000 for a rubber belt if it can be depended upon to last more than sixteen months; but if the average life of the rubber belts on the conveyor is no more than fourteen months, the canvas belt is more economical.

Canvas belts are sold under trade names that tell nothing about the make-up of the duck, the stitching, the nature of the waterproofing compound, the amount of stretch taken out, or the time of seasoning or curing. Correct knowledge on these points is not widely disseminated. Some of the failures of canvas belts in service must be attributed to faulty methods of manufacture, but disappointments in service are sometimes due to using the wrong kind of belt. A canvas belt bought from a jobber's stock and run over troughing idlers may be a failure as a conveyor belt, although it might have been a good transmission belt. Conveyors are not all alike, and a belt suited to one may last only a short time on another; good elevator belts are not necessarily good conveyor belts. Canvas belts are naturally denser and stiffer than rubber belts and in the narrower widths do not trough so well on idlers designed for use with rubber belts.

Canvas belts are not recommended for the difficult work of conveying or elevating sharp ore in the presence of water. Oil-treated belts with Class 1 saturation are not sufficiently waterproof. Belts treated with Class 2 compounds are better in that respect, but for continual stretching and bending while exposed to water, as in a wet elevator handling mineral pulps, they do not resist the water so well as a good rubber belt or a balata belt. They are also deficient in resistance to cutting and abrasion in the presence of water as compared with other belts.

Much of what has been said in the preceding paragraphs about canvas belts applies to balata belts also. They are stronger and stiffer than rubber belts and in the narrower widths do not trough so well.

Strength of Belts. In a rubber, canvas, or balata belt the ultimate strength depends largely on the strength of the duck of which it is made; in a solid-woven belt the strength varies according to the sizes of the threads and the closeness of the weave. Belt ducks are generally referred to as weighing so much per square yard, or per yard of length 42 inches wide, and, in a general way, the heavier the duck, the stronger it is. There are, however, many different ways of assembling warp

threads and filler threads to make a duck weigh so many ounces per square yard; a duck for a canvas belt may have a filler relatively heavier than the same weight of duck made for a rubber belt. The duck for a balata belt can be more closely woven than a rubber belt duck. It is therefore not possible to compare the strengths of belts solely on the basis of the weights of the duck.

There is no direct connection between the strength of a finished belt and the strength of the duck as tested in a testing machine. The plies do not all take an equal share of the load, nor is the load uniformly distributed to all the warp threads in the width of the belt. The "friction" or the impregnation of a belt has an important influence on its strength. In a rubber belt, the layers of "friction" rubber act as a support for the threads, prevent distortion of belt structure under load, and help to distribute the load among the plies of duck. In a stitched canvas belt, the oxidized or gelatinized oil which fills the spaces among the threads acts in a similar way, but with less effect, because it is not so strong as the rubber. In a balata belt, the saturating gum is stiff and strong, and with a given structure of duck it makes a belt higher in tensile strength and less in stretch than other means of holding the plies together. In the best impregnated solid-woven belts, the structure of fabric is quite dense and the strength is high, but in most plain, white, solid-woven belts the weave is not so close, the threads are not kept in place by an impregnating gum, and consequently the stretch is greater and the breaking strength is less. Table 3-K records tests of two solid-woven belts not impregnated and one stitched canvas belt with Class 1 impregnation.

Strength of Rubber Belts. Since no two makes of rubber belts are alike in the weave of the duck even for the same nominal weight, it is not possible to assign definite strengths to the belts, considering also the contingencies of manufacture. It is really not necessary that belts should be rated by their breaking strengths; as is pointed out in Chapter 6, belts are seldom strained to more than one-tenth of their ultimate strength, and with many belts, other factors determine their suitability to particular service and their life.

As a guide for determining the allowable working tensions, the following may be taken as representing the average breaking strengths of rubber belts as made in this country, measured per inch of width per ply of thickness:

Belts made of 28-oz. duck	280 lb.
Belts made of 30- or 32-oz. duck	300 lb.
Belts made of 36-oz. duck	325 lb.
Belts made of 42-oz. duck	375 lb.

Strength of Fabric Belts. Stitched canvas belts of 32-ounce duck will break at about 300 pounds per inch per ply.

Balata belts are made of various weights of duck; tests of the best grade using 38- or 40-ounce duck show about 400 pounds per inch per ply. In general they are about 20 or 25 per cent stronger than rubber belts of 28- or 32-ounce duck.

On the relation between ultimate strength and working tensions see Chapter 6.

Belt Fasteners. A fastener for a belt joint must be strong and yet so flexible or so short that it will bend around the pulleys without breaking the belt by bending it crosswise or without pulling it apart by tearing out the cut ends. A fastener for a conveyor belt must also be flexible crosswise or be applied in short sections so that the belt can conform to the contour of the troughing idlers.

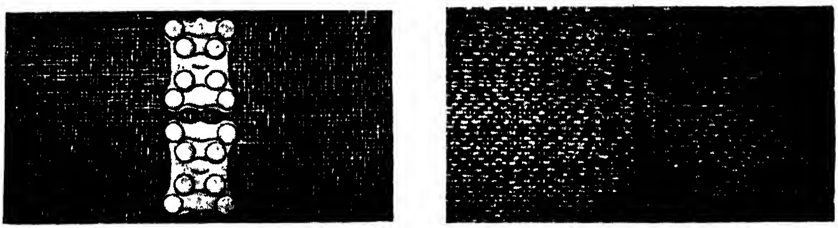


FIG. 3-23. Belt Joint with Crescent Plates and Crescent Rivets.

Rawhide lacing and the coiled wire lacing used for transmission belts are not used for conveyor belts. They do not make a closed joint, and as applied to fabric belts they are objectionable because they transmit the pull to the filler or crosswise threads in the belt, and these are likely to pull out when the lacing holes are pierced close to the cut ends of the belt. To avoid pulling out and to transmit the pull to the warp or lengthwise threads, a fastener for a fabric belt should grip the warp threads by squeezing them together and preferably by clinching them in some way.

Fasteners for Conveyor Belts. Several styles of metal fasteners are used for conveyor belts.

1. Steel plates with split rivets (Fig. 3-23) are made by the Conveying Weigher Company, New York; Bristol Company, Waterbury, Conn.; Crescent Belt Fastener Company, New York, and others. Except in very thick belts, it is not necessary to punch holes for the rivets; the rivets, when driven through the belt, compress the warp threads and clinch around them.

2. Clinch hooks are metal plates with projecting prongs which are driven through the belt with a hammer, then clinched on the pulley side.

The Talcott fastener (Fig. 3-24) (W. O. & M. W. Talcott, Providence, R. I.) is made of malleable iron; the prongs are stiff in the direction of the belt pull, are tapered both ways, and have rounded corners so as not to cut the fibers of the belt. The points are thin and

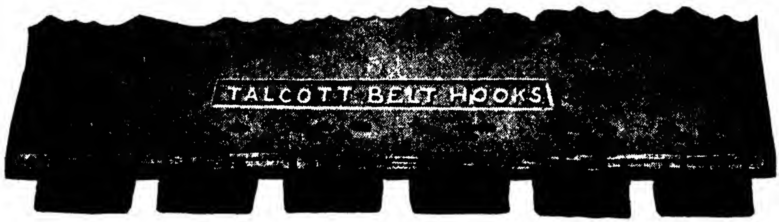


FIG. 3-24. Talcott Belt Fastener.

are easily clinched into the thickness of the belt. The backs of the fasteners are stiff and strong.

The Bristol fastener (Fig. 3-25) is of sheet steel; the tapered prongs enter between the warp threads and get a firm hold by wedging them together.

All these fasteners are made in short sections and should be applied to the belt so as not to interfere with the troughing.

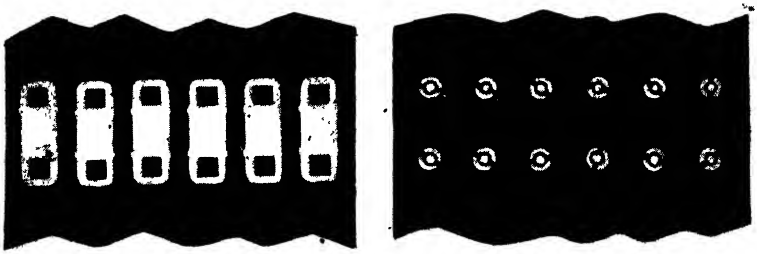
To prevent leakage of fine stuff at the joint, a piece of duck or thin belt is sometimes placed under the fasteners.

3. Bolted fasteners are steel plates with bolts. The Flexco Belt Fastener (Flexible Steel Lacing Company, Chicago) (Fig. 3-26) is used only for belts ⅜ inch or more in thickness that run over pulleys at least 2 feet in diameter. It does not leave the belt flush on either side and hence there is noise when it passes over idlers, but it is a strong fastener for very heavy belts. It requires the belt to be drilled for the bolts, but these take no shear; the hold depends upon the powerful compression between the opposite plates and the wedging action of the conical nuts. The Jackson fastener is similar in principle, but it has oval cup washers on the pulley side of the belt, and that side of the joint is smooth and runs quietly over the idlers. It is more fully described as a fastener for elevator belts. (See page 358.)

FIG. 3-25. Bristol Steel Belt Fastener.

Flexco Rip Plates are wider than Flexco fasteners and are used to mend longitudinal rips and make patches.

4. Hinge-pin Fasteners. "Alligator" lacing (Flexible Steel Lacing Company) consists of sections of sheet steel with prongs driven into the belt from both faces and made to interlock over a steel or rawhide pin to form a hinge between the butt ends of the belt. It is well suited to belts run flat and for package conveyors where a close butt-joint is not essential. The fastener is thin, lies close to the belt, and is not likely to damage packages at loading or discharge stations.



Pulley Side of Belt Showing Steel Plates with Square Heads of Bolts Set Flush.

Upper Side of Belt Showing Steel Plates and Conical Nuts.

FIG. 3–26. Flexco Belt Fastener for Heavy Belts.

For recent foreign fasteners of this type see page 278.

Sizes of Fasteners. In using any metal fastener, be guided by the maker's instructions as to the way to apply it and the proper size to use for the particular width and thickness of belt. Some kinds suitable for leather belts make a poor joint in a fabric belt; the prongs or rivets are too close to the cut ends of the belt, and the threads pull out. In using steel plates with split rivets, the rivets must be of the proper length for the thickness of belt.

A joint made by a metal fastener is never as strong as the body of the belt, and in a fabric belt it is a point of weakness because it opens the way for the entrance of moisture. In important work, the cut ends of the belt are covered with several coats of rubber cement before the fasteners are put on. This hinders water from getting into the belt.

Vulcanized Splices. Rubber transmission belts are sometimes made endless at the factory by splicing the ends and vulcanizing the joint, but conveyor and elevator belts are seldom, if ever, made that way because of the difficulty of getting an endless belt in place over the

pulleys and idlers. Splices can be vulcanized with the belt in place by using a portable vulcanizing press. James C. Heintz & Company, Cleveland, Ohio, make these presses in sizes from one weighing 50 pounds up to one that weighs 3200 pounds and will take a 60-inch belt. Fig. 3-27 shows such a press in use. The upper and lower platens are electrically heated and are mounted in frames that can be clamped together by bolts. After the two ends of the belt have been cut back a ply at a time and fitted together, they are cleaned, cemented, allowed to dry, fitted again, cemented, put in the press, and kept under

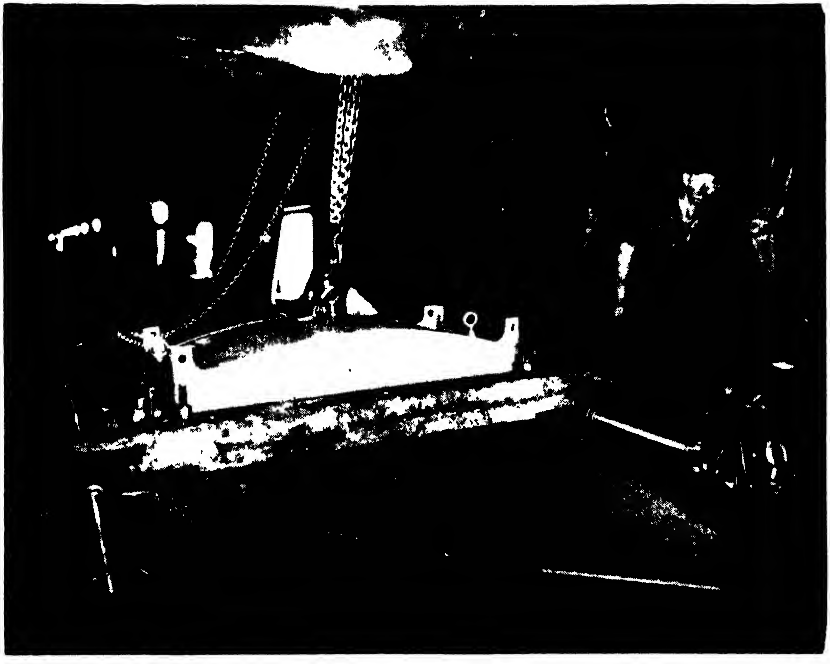


FIG. 3-27. Splicing a Wide Belt with a Portable Vulcanizing Press.

proper pressure and temperature for about half an hour. Directions for preparing the ends of the belt and making the splice are given in publications issued by the makers of the press and by makers of rubber belts. Making a good vulcanized splice requires skill and care. In plants that use many belts it pays to buy a press and train men to use it, but for an occasional job it is better to have one of the rubber belt companies bring its own press equipment to the place and do the work. Most of the companies keep the equipment on hand in the principal agencies. Large portable vulcanizers for the widest belts are gen-

erally kept at the factory. To splice a 48-inch belt within, say, 500 miles of Akron, the cost would be about as follows:

1 man, 3 days, average @ $12.00	$ 36
Traveling expenses, about	50
Living expenses, about	15
Transportation of equipment, about	30
Material	3
Miscellaneous	10
	$144

That is, the cost would probably be between $125 and $150.

One company that uses many 48-inch conveyor belts has always used vulcanized splices. In the original installation there were over 100 such splices; experience has shown that they are fully 90 per cent

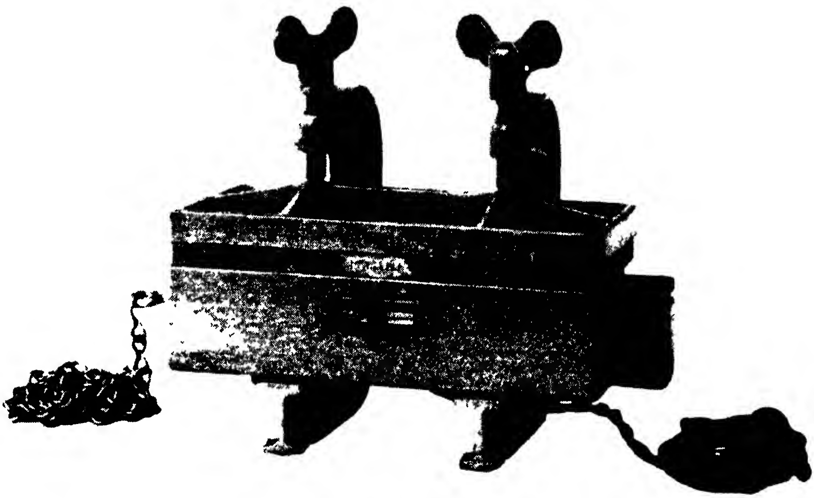


FIG. 3-28. Single Platen Vulcanizer. (James C. Heintz and Company.)

as strong as the belt, they outlast any splice made with metal fasteners, and they give less trouble. The company's system of belt inspection is complete and thorough; each belt is inspected every day by a foreman and his assistant. For two full revolutions of the belt, a light is held close to the pulley side of the belt; if there is a tear anywhere or a loose ply at a splice, the belt is stopped and temporary repairs are made with metal fasteners. Permanent repairs are made with a vulcanizer by men on the night shift. If the damaged spots are small, a single-platen vulcanizer (Fig. 3-28) is used.

The following is adapted from handbooks issued by the Goodyear

Tire and Rubber Company and by the B. F. Goodrich Company. Probably no improvement in conveyor belt practice has done more to save maintenance time and expense than the perfection of the field-vulcanized splice. It is much stronger than any joint made with metal fasteners, lasts much longer, causes no pounding on the return idlers, and makes no holes in the belt through which fine stuff or water can enter and damage the fabric. It will not tear apart suddenly, but will give warning by showing a partial separation of the plies at the splice in time for a repair to be made. It is flush on both sides; the belt can be cleaned better because there is nothing to strike against a scraper or a cleaning brush; a cleaner belt means less drip of dirt on the return run and less accumulation of dirt on the rims of the return rolls—dirt which when it hardens may cut a belt or cause it to run crooked.

Vulcanizing Repaired Spots. If the cover of a rubber belt is damaged by a cut, a gouge, or a tear, moisture and dirt are likely to enter the carcass and rot the duck. It always pays to repair such wounds promptly; the best way is to cut out the bad spot and vulcanize new rubber in its place. This can be done with a splice vulcanizer or with a smaller one intended for repairs only. Fig. 3-28 shows an electric vulcanizer with a single platen about 5 inches by 11 inches that costs about $50. It can repair a larger area by vulcanizing one spot and then shifting the vulcanizer until the whole area has been covered.

Pamphlets issued by J. C. Heintz & Company and by rubber-belt manufacturers describe minutely and illustrate with pictures each step in belt repairing and splicing.

Sandvik Steel Belt. This belt, made at Sandviken, Sweden, is a band of 0.65-carbon cold-rolled steel, 0.03 or 0.04 inch thick, hardened and tempered. It comes in widths up to 32 inches and in lengths up to 350 feet. It is possible to make belts up to 6 feet or even more in width by assembling two or more bands with longitudinal riveted joints. The band, whatever the width, cannot be troughed but must be run flat. The loaded side may be supported on rollers spaced 4 feet or more apart, the distance depending on the weight of the load on the belt. Fig. 3-29 shows the cross section of a 16-inch belt distributing tempered foundry sand. The flanged guide idlers shown are spaced about 50 feet apart and are intended only as safeguards. Normally the belt is made to run straight by controlling the tension in the steel band. The return run is carried on rollers set 10 or 15 feet apart. If the material is not abrasive, the carrying run may be supported by wooden strips as in the sugar conveyor shown in Fig. 3-30.

The carrying capacity of the belt depends on how the material piles, and whether it flows freely or not, but as compared with a fabric

belt run flat the Sandvik belt can carry more because it works under a greater tension, the sag between the idlers is less, and the disturbance of the load in passing over the idlers is less also. A 16-inch Sandvik

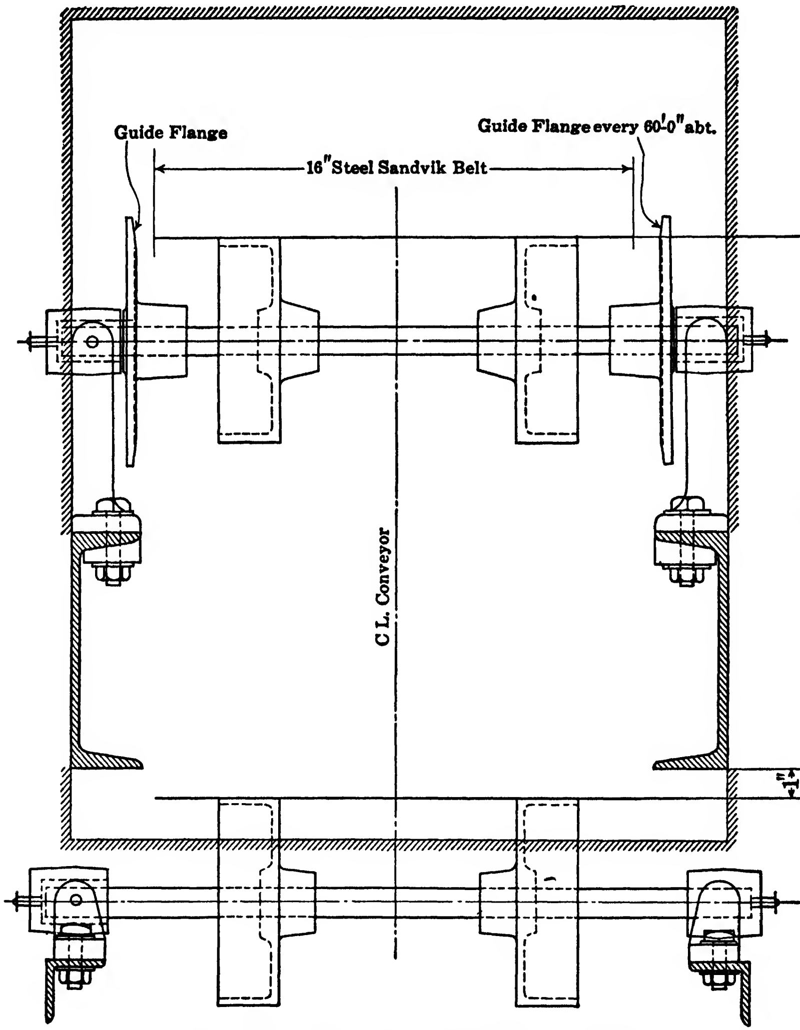


FIG. 3-29. Steel Belt Supported by Rollers.

belt is rated at about 0.1 cubic foot per foot of belt for coal or similar materials, but more for chemicals that do not flow so readily, and up to 0.2 cubic foot per foot for raw sugar and other materials that can form a high pile on the belt. For some non-abrasive materials, the belt, if not

too long, may be used with continuous skirt boards so that it forms the moving bottom of a rectangular trough, in which case it may carry three or four times as much as a belt without skirt boards. See Fig. 3-31.

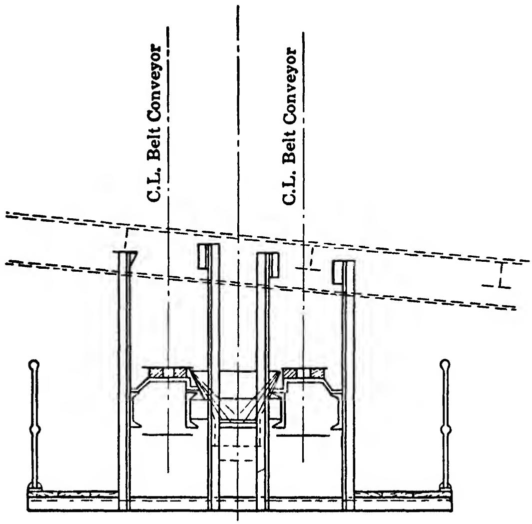


FIG. 3-30. Steel Belt Sliding on Wooden Strips.

The normal speed of the steel belt for conveyors more than 130 feet centers is 200 to 300 feet per minute; for conveyors shorter than 70 feet, where the riveted lap joint in the belt goes oftener around the end pulleys, the speed should be less than 150 feet per minute. If the belt slides, the speed should not be more than 200 feet per minute.

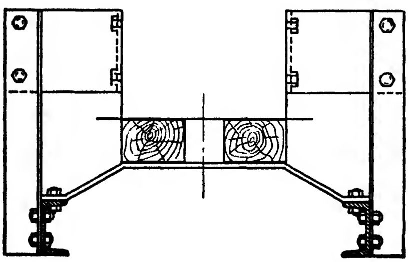


FIG. 3-31. Sliding-trough Conveyor.

The makers of the belt recommend 40-inch end pulleys for belts 16 inches or more in width but they permit the use of smaller pulleys for narrower belts. The pulleys are made with heavy arms and rims; the face is narrower than the belt and is turned with a crown that has a flat bearing on the belt for about $\frac{4}{10}$ of its width. If the conveyor is long or hard-worked, it is permissible to cover the drive pulley with wood lagging. It is important to keep stray pieces of material from getting between the belt and the pulleys; for that purpose, deflectors

are set diagonally over the return run, and scrapers are used to keep the pulley rims clean; otherwise the band might be dented.

The belt can take its own feed through an opening in the bottom of a hopper; but in handling lumpy material, care must be taken to avoid a direct drop to the belt. It is not necessary to feed with velocity in the direction of travel.

One of the advantages of the Sandvik belt is the ease of discharge at various points along the run by means of traveling or stationary plows. The plow consists of an arm, usually adjustable as to angle and position, carrying a number of short, flexible scraper blades, lapped shingle-fashion. These blades bear on the carrying surface of the belt, scrape it clean, and discharge the material over the edge. Since they can be set to remove all or any part of the load, it is easy to have a simultaneous discharge at any desired number of places.

The Sandvik belt is guided by the proper alignment of the end pulleys and of the supporting rollers, the rollers being generally mounted with bearings having lateral adjustment. Since the belt does not stretch from wear in service, no take-up is necessary other than some means to allow for expansion and contraction due to change of temperature. The splice is a lap joint with one row of $\frac{3}{16}$-inch rivets.

The Sandvik belt has been used in Europe for thirty years and there are several thousand installations in Germany, Sweden, and England. A number of these belts have been sold in this country, for handling clinker, phosphate rock, sugar, clay, foundry sand, and some wet sticky materials that cannot be discharged clean from fabric belts, also some chemicals that are injurious to rubber or cotton. As between a fabric belt discharged through a tripper, and a Sandvik belt discharged by means of plows, the advantage frequently lies with the latter.

Some makers of chemicals have found it possible and advantageous to use Sandvik belts as part of continuous production processes. The chemical product while still hot or in plastic form is fed upon the moving belt and cools and hardens in transit while being conveyed from one point to another in the process of manufacture.

Fig. 3-32 shows three parallel 32-inch Sandvik belts used in England in a cake bakery. The dough is cut by machine, deposited on the bands, carried at a controlled speed through an oven, baked, then cooled. The finished cakes are scraped off, then packed. Packing plants in the United States use Sandvik belts for carrying meat, fish, oysters, bacon, and other food products. The belt imparts no taste or odor to the food, is not hurt by them, and can be washed with hot water or scalded with steam without injury.

Sandvik belts are light, run quietly, and take no more power than fabric belts, sometimes less power. The makers issue catalogs and bulletins giving information about horsepower, accessories, and carrying capacities. Each installation is an engineering job and should be referred to the makers or their agents for advice. (New York representative, 111 Eighth Ave., New York. British agency, Dawlish Road, Selly Oak, Birmingham, England.)

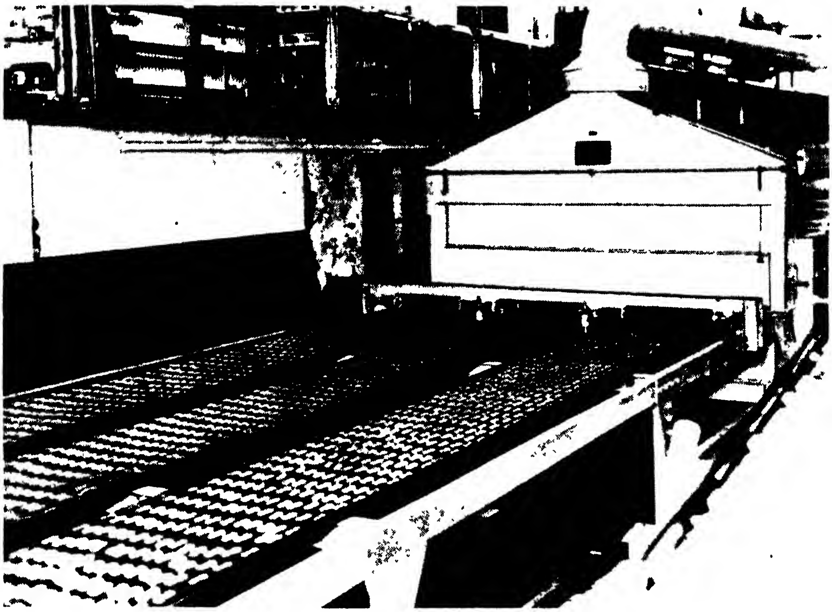


FIG. 3-32. Three Steel Belts, Each 32 Inches Wide, at the Discharge End of a Continuous Bake Oven.

The Bergh patent of 1938 discloses the idea of cold-rolling a steel band for a conveyor so that its normal shape is concave on one side and convex on the other, like the steel tape used in some measuring rules. The concavity of such a steel band will permit it to carry more material than a flat band will carry, yet the band is to be flexible enough in cross section so that it will flatten down in passing around the end pulleys.

Steel belts similar to the Sandvik are made in Germany by Hoesch of Dortmund.

Wire mesh belts have been used in Europe for years for general conveying purposes, but in this country they are limited in general

to carrying materials through processing operations. They are coming more and more into prominence for washing and drying, cooking and cooling, heat-treating and annealing, and in many other operations where, because of heat or wet, fabric or rubber belts are unsuitable. The metal of the belt can be of steel, brass, copper, or any of the alloys which resist corrosion or great heat. Their open structure permits

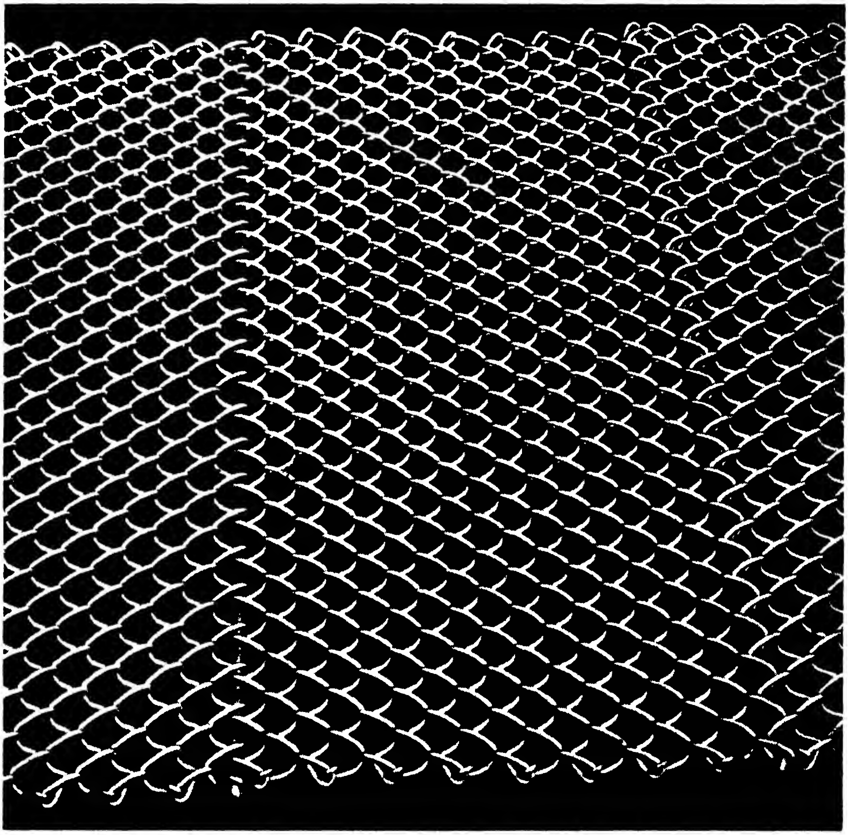


FIG. 3-33. Woven Wire Belt with Reinforcing Rods.

liquid to drain away from the material carried on the belt and allows free circulation of air and heat. In glass-annealing ovens the wire mesh apron may be as wide as 12 feet, and it will withstand temperatures up to 1600° F.

The simplest form of such a belt is an assemblage of flattened helical coils of wire interlaced to form a flexible band which can wrap around the end pulleys of a conveyor and be driven by frictional

contact with one of the pulleys. To make a belt of greater strength and crosswise stiffness, cross wires or rods may be woven in to act as hinge pins and spreaders (Fig. 3-33), and where the pull in the conveyor requires it, the edges of the belt can be reinforced by interwoven wires, or by sprocket chains.

There are many United States and foreign patents on methods of strengthening edges of wire belts; turning them up as flanges to keep material from spilling off, attaching plate flanges, making wide belts by assembling several narrower belts, using a fine-mesh belt as a cover

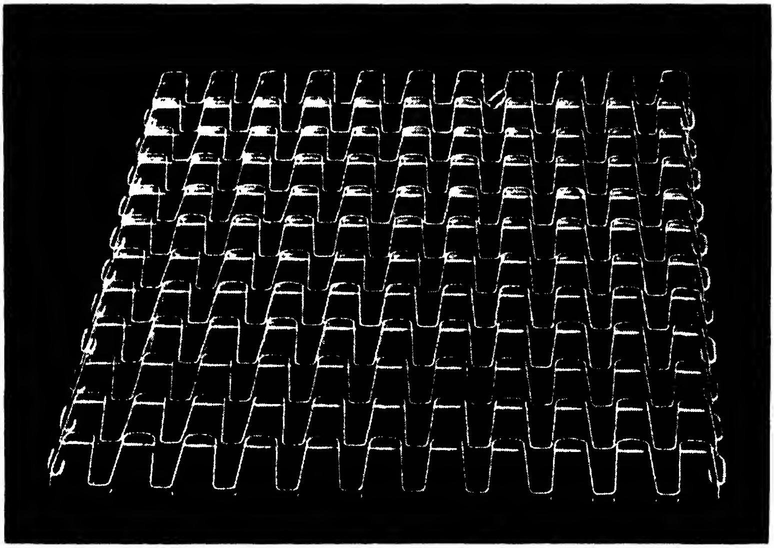


FIG. 3-34. Conveyor Belt of Flat Steel Strips with Hinge Pins.

for a coarse-mesh belt to form a smooth carrying surface, making up belts in sections with alternating right-hand and left-hand weave, using, alternately, right- and left-hand coils with crimped hinge pins.

Another form of steel belt (Fig. 3-34) is made of flat metal strips bent into the form often seen in doormats, and joined by through-pins or wires which form hinges for the sections.

The following information for the operation of wire mesh belts is given by the Wickwire Spencer Steel Company, New York:

1. In the use of metal conveyor belts any construction of the conveyor frame and supports which may cause the wires to bend will decrease the life of the belt.
2. Make certain flexure of the belt takes place in the joints instead of within the wire.

3. Flanges on pulleys will not control side travel of a woven wire belt, and they will cause early failure on account of flexure in the metal.
4. Wire belts do not require crowned pulleys and should always be operated over straight face pulleys.
5. Alignment of all pulleys should be adjustable and side travel of the belts controlled by adjustment of the bearings. A wire belt always travels to the low side of the pulley.
6. Whenever possible, small pulleys should be avoided. They cause excessive wear and increase tendency to side travel.
7. The friction coefficient of wire belts on smooth steel or iron pulleys is approximately $\frac{3}{10}$ at room temperature. Traction may be increased by covering the drive pulley with material having a high coefficient of friction. (By increasing the traction, the tension required to drive is reduced and in consequence the life of the belt increased.)
8. Pulleys having a rubber covering, vulcanized to the surface, make the best drives where such a surface is not subject to high temperatures.
9. Supports for wire belts whether revolving or stationary, should extend the full width of the belt. Do not support metal belts along the edges only.
10. Wire conveyor belts should be kept free from grit or abrasive material that may wear the joints.
11. A wire belt showing excessive wear on one side may be turned over, thus doubling its period of usefulness.
12. Before tension is applied to a belt, whether new or used, its entire length should be examined to make certain that no spirals have become turned up on edge.
13. Take-ups on high-temperature belts should slide freely to allow for expansion and contraction of the belt, and the alignment should be constant, regardless of temperature.
14. Avoid overheating of high-temperature belts by frequently checking temperature controls.
15. Periodical checking of metal conveyor belts and minor repairs when necessary will prolong the life of the belt.

To these may be added the following:

16. If the return run of the belt is supported on idlers or rollers, do not place them too far apart.
17. Run the belt with as little take-up tension as possible.
18. Before you buy the belt, make sure that the manufacturer understands fully and clearly how it is to be used; after you get the belt follow his directions for putting it in place and using it.

Generally the maker of the wire mesh belt furnishes the belt only; the purchaser buys the shafting, pulleys, idlers, and driving machinery from others.

Several forms of construction intended to combine the advantages of fabric or rubber belts with those of wire mesh belts have been patented, but they have not come into actual use. Fig. 3-35 shows a wire mesh structure imbedded in rubber and provided with flexible hinge pins so that the belt could be troughed (Pattee patent, 1916). A wire mesh belt with a filling of jute or some such fibrous material was patented by Kida in 1918. Such devices are likely to fail because the

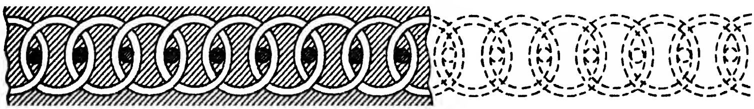


Fig. 3-35. Wire Mesh Imbedded in Rubber.

filler material will fall out when it disintegrates from the repeated bending where the wire coils hinge on each other.

Wire mesh belts have been used abroad to handle bulk materials; to prevent leakage of fine stuff, thin strips of wood or metal have been inserted in the flattened coils, or the top surface of the belt has been covered with thin fabric. These devices must be regarded as makeshifts. They are heavy and clumsy, they do not prevent leakage altogether, and there is always spill of material which clings to the return run of the belt.

CHAPTER 4

SUPPORTING AND GUIDING THE BELT

Chapter 2 tells why belt conveyor idlers with ball or roller bearings have been found to be better than old-style idlers with plain bearings and have gradually displaced them. Ball-bearing idlers had been made in a small way before the introduction of the all-steel unit roll shown on page 27. In most of them the pulleys were of cast iron bored and fitted with a ball bearing at each end of the hub. They required a grade of shopwork better than that usually put on belt idlers; they were made in small lots and cost too much. After the unit-roll type had been in successful use for some years, designs of other kinds of idlers were improved, costs were cut, prices reduced, and now idlers with ball bearings or roller bearings are sold by American manufacturers at prices that compete with old-style idlers with plain bearings. For belts wider than 30 inches, three-roll idlers of modern design cost no more than old-style five-pulley idlers and may cost even less. As to the carrying capacity of belts troughed over them there is little or no difference between five pulleys and three pulleys, but the idlers with three pulleys or rolls have the advantage of fewer parts and easier lubrication.

The success of the three-roll carriers on the Colonial Dock conveyors of the H. C. Frick Coke Company (see page 97) put an end to doubts about using them for wide belts, and since 1924 the trend in the United States has been toward three-roll carriers for all widths of conveyor belts. It must be said, however, that five-roll troughing idlers are preferred by some engineers for wide belts that carry large quantities of heavy ore and stone, especially if the load is piled very deep on the belt. In Great Britain the five-roll idler is used for general conveyor work although the three-roll idler is more common for mining work underground. In Germany the three-roll idler has been made the "norm," that is, the standard. It is shown on page 92.

Idler Assembly. Fig. 4-1 shows a typical American troughing idler. A cross member which may be a channel, a tee, or an angle carries four brackets, usually of malleable iron, with slotted top ends. These slots

take the ends of the roll shafts, keep them from turning, and permit the roll assembly to be lifted out when necessary.

Table 4-A gives dimensions of such an idler together with the corresponding return idler.

When the rolls of a troughing idler are mounted on a stand the easiest way to take the end thrust in the inclined rolls is to fit them with tapered (Timken) or concaved (Shafer) roller bearings and run them on fixed shafts. In some idlers ball bearings capable of taking end thrust as well as radial load have been used, but they are not so

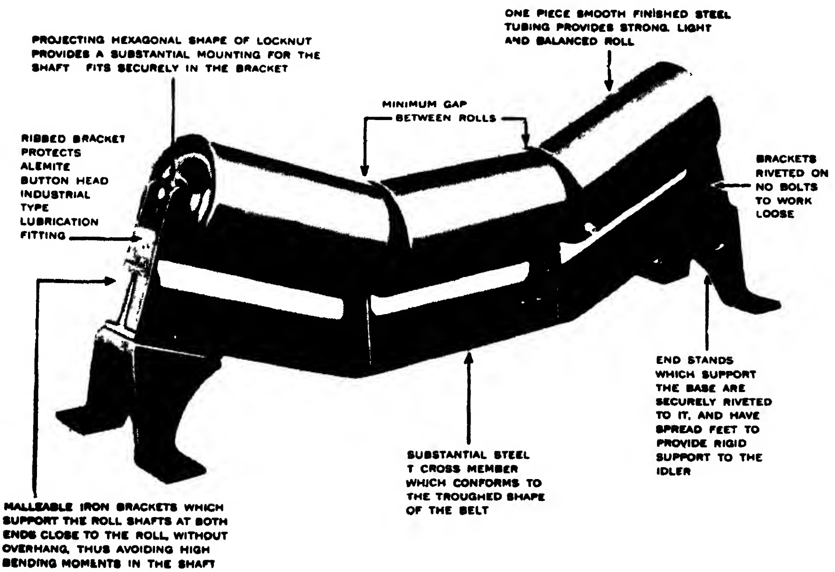


FIG. 4-1. Typical Anti-friction Troughing Idler. (Link-Belt Company.)

common. The dead-shaft arrangement where the shafts do not rotate makes the roll construction and idler assembly simpler than with the live-shaft construction in which the roll is tight on the shaft and the shaft revolves.

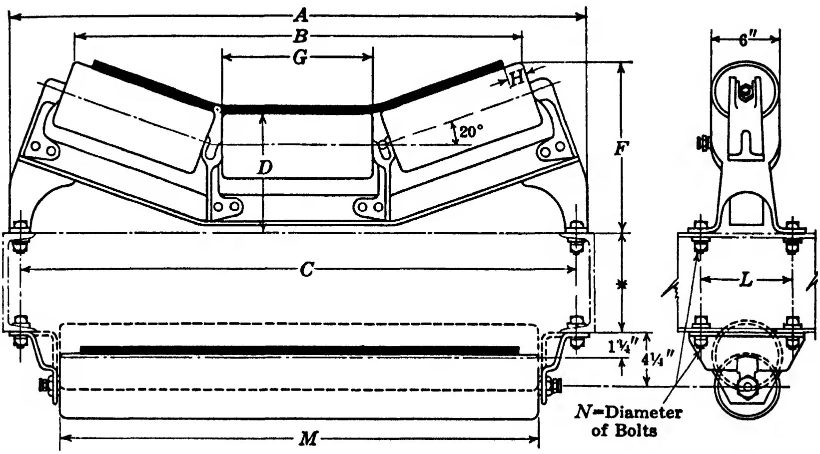
Troughing idlers are made in many styles according to the work expected of them. Fig. 4-2 (Stephens-Adamson Manufacturing Company) shows an idler of the kind mentioned on page 27 with sheet-steel stand and sheet-steel ball-bearing pulleys. It is for a narrow belt and weighs about 50 pounds. Alongside it in the illustration is an idler for a 60-inch belt on an ore conveyor. All parts are cast steel; it weighs 900 pounds.

A five-roll idler for heavy work is shown in Fig. 4-3, Robins Conveying Belt Company.

TABLE 4-A

TYPICAL AMERICAN TROUGHING IDLER AND RETURN IDLER

Link-Belt Company



Dimensions in Inches

Width of Belt	A	B	C	D	F	G	H	L	M	N
14	$24\frac{7}{8}$	$16\frac{9}{16}$	23	$9\frac{1}{2}$	$11\frac{5}{16}$	$5\frac{9}{16}$	$1\frac{5}{8}$	$4\frac{1}{2}$	$16\frac{7}{8}$	$\frac{1}{2}$
16	$26\frac{3}{4}$	$18\frac{5}{16}$	25	$9\frac{1}{2}$	$11\frac{1}{2}$	$6\frac{3}{16}$	$1\frac{9}{16}$	$4\frac{1}{2}$	$18\frac{7}{8}$	$\frac{1}{2}$
18	$28\frac{3}{4}$	$20\frac{1}{8}$	27	$9\frac{1}{2}$	$11\frac{3}{4}$	$6\frac{13}{16}$	$1\frac{1}{2}$	$4\frac{1}{2}$	$20\frac{7}{8}$	$\frac{1}{2}$
20	$30\frac{3}{4}$	$22\frac{1}{4}$	29	$9\frac{1}{2}$	12	$7\frac{9}{16}$	$1\frac{5}{8}$	$4\frac{1}{2}$	$22\frac{7}{8}$	$\frac{1}{2}$
24	$34\frac{3}{4}$	$26\frac{1}{4}$	33	$9\frac{1}{2}$	$12\frac{7}{16}$	$8\frac{15}{16}$	$1\frac{11}{16}$	$4\frac{1}{2}$	$26\frac{7}{8}$	$\frac{1}{2}$
30	41	32	39	$10\frac{1}{4}$	$13\frac{7}{8}$	$10\frac{15}{16}$	$1\frac{5}{8}$	$7\frac{1}{2}$	$32\frac{7}{8}$	$\frac{5}{8}$
36	47	38	45	$10\frac{1}{4}$	$14\frac{5}{8}$	$12\frac{15}{16}$	$1\frac{5}{8}$	$7\frac{1}{2}$	$38\frac{7}{8}$	$\frac{5}{8}$
42	$53\frac{11}{16}$	$44\frac{5}{8}$	51	$10\frac{1}{4}$	$15\frac{3}{8}$	$15\frac{1}{4}$	$2\frac{1}{8}$	$7\frac{1}{2}$	$44\frac{7}{8}$	$\frac{5}{8}$
48	$59\frac{11}{16}$	$50\frac{7}{16}$	57	$10\frac{1}{4}$	16	$17\frac{1}{4}$	$2\frac{1}{8}$	$7\frac{1}{2}$	$50\frac{7}{8}$	$\frac{5}{8}$
54	65	$56\frac{1}{16}$	63	$10\frac{1}{4}$	$16\frac{13}{16}$	$19\frac{5}{16}$	$2\frac{1}{4}$	$7\frac{1}{2}$	$56\frac{7}{8}$	$\frac{5}{8}$
60	71	62	69	$10\frac{1}{4}$	$17\frac{1}{2}$	$21\frac{5}{16}$	$2\frac{1}{4}$	$7\frac{1}{2}$	$62\frac{7}{8}$	$\frac{5}{8}$

Fig. 4-4, Chain Belt Company, shows an idler designed for low headroom; the end rolls are inclined at 30° to keep material on the belt.

Fig. 4-5, Stephens-Adamson Manufacturing Company, illustrates an idler for use in mining work underground where light weight (for portability) and shallow depth (to clear low roof) are important con-

siderations. The three-roll shafts are clamped in malleable iron supporting stands to form a stiff upper chord, and a bent rod with

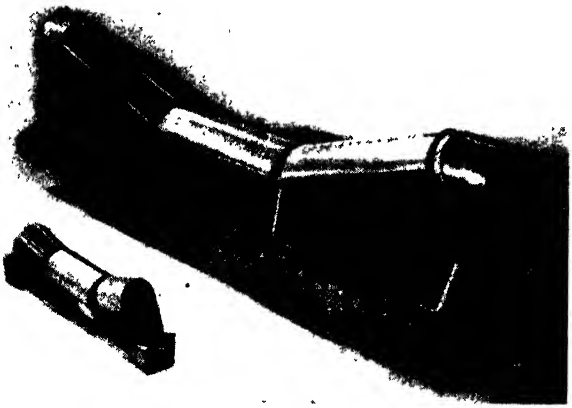


Fig. 4-2. A Large Idler and a Small One. (Stephens-Adamson Manufacturing Company.)

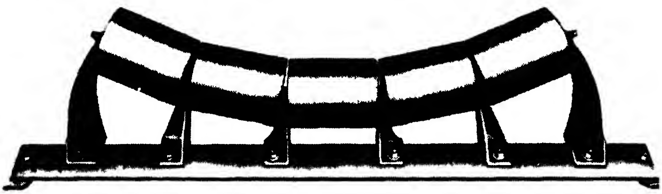


Fig. 4-3. Five-roll Troughing Idler with Two Places for Lubrication. (Robins Conveying Belt Company.)

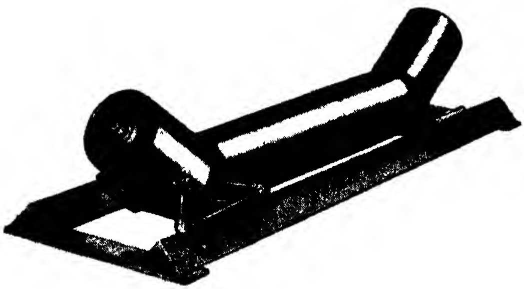


Fig. 4-4. Idler Designed for Low Headroom. (Chain Belt Company.)

pipe spacers and nuts forms the lower chord of a truss frame. Troughing idlers of this kind weigh about half as much as the kind shown on page 88, but they have some features which limit their field of use.

Robins patent 2,031,618 of 1936 refers to a three-roll idler in which the rolls are not mounted on through shafts, but on short stub shafts

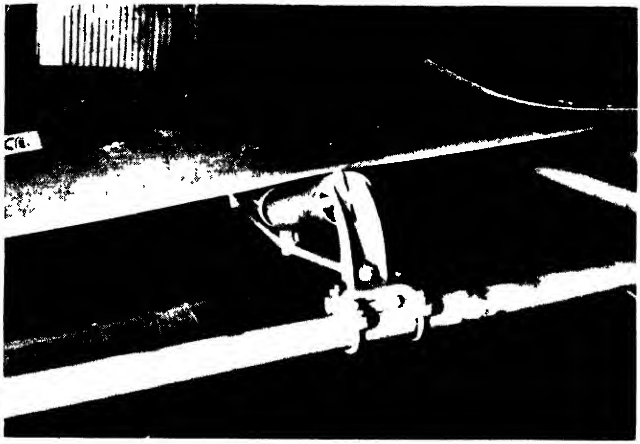


FIG. 4-5. Pipe Frame Troughing Idler. (Stephens-Adamson Manufacturing Company)

which project from the stands on which they are clamped. The short shafts are turned with the ends ball-shaped and run in closed-end ball

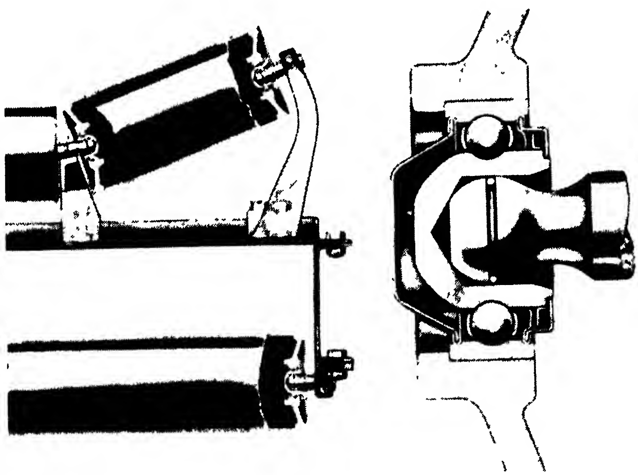


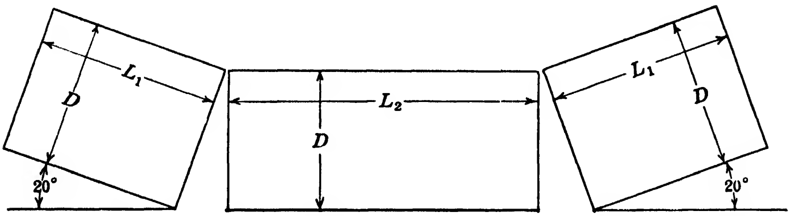
FIG. 4-6. Idler Rolls without Through Shafts; Closed-end Bearings.

bearings mounted in the ends of the rolls. Fig. 4-6 shows the assembly. Some have been built. See page 101.

Table 4-B shows German standard troughing idlers. Rolls are of seamless steel tubing. Ball bearings or roller bearings are coming into use, but are not general. Lubrication is by grease.

TABLE 4-B

GERMAN STANDARD TROUGHING ROLLS



Dimensions in millimeters

Width of Belt	For Light Work		For Heavy Work		Length of Roll	
	D	Thickness of Tube Wall	D	Thickness of Tube Wall	L_1	L_2
400	108	3 75	133	4	165	165
500	108	3 75	133	4	165	270
650	108	3 75	133	4	250	250
800	108	3 75	133	4	250	445
1000	133	4	159	4 5	380	380
1200	133	4	159	4 5	380	635
1400	133	4	159	4 5	530	530
1600	159	4 5	191	5 5	530	735
1800	159		191	5 5	665	665
2000	159		191	5 5	665	870

Live Shafts; Dead Shafts. If the outer race of a ball bearing or a roller bearing is tight in the idler roll, and the inner race revolves with the shaft, the balls or rollers travel at a lower speed than if the shaft is fixed as is usual in troughing idlers. In Fig. 4-7, showing a ball bearing on a $\frac{3}{4}$-inch shaft, the diameter of the inner track of the balls is about $1\frac{1}{16}$ inch and that of the outer track $1\frac{13}{16}$ inch. The speeds of travel of the balls for the two conditions are therefore in the ratio of about 1 to 1.7, and the friction losses *within the bearing* might be in that proportion. This difference in favor of the live shaft, however, is very small; a greater difference is that with the dead-shaft construction

the stationary shaft is surrounded by a mass of grease which may exert a measurable "drag" on the revolving roll. On the other hand, in the usual idler with dead shafts, the accurate alignment of the two ball or roller bearings in each roll is simple and permanent; it depends only on boring and facing operations in a machine shop and is independent of any inaccuracy or distortion of the supporting frame of the idler. But in the live-shaft construction the two bearings which carry each shaft must be accurately aligned on the idler stand or else mounted in such a way as to allow a certain amount of self-adjustment.

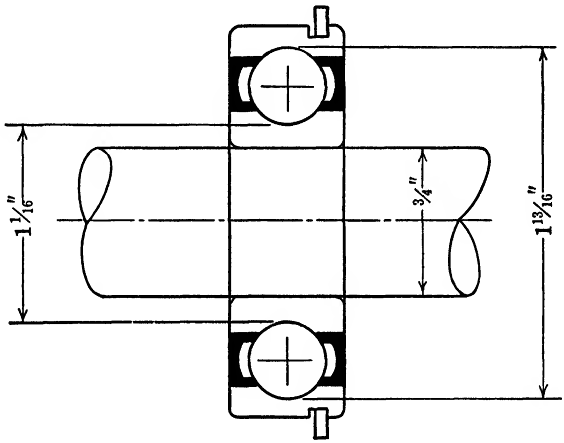


FIG. 4-7. Relative Diameters of Inner and Outer Ball Tracks.

For these reasons and because it is cheaper, the dead-shaft construction is the one generally used.

For an idler with live shafts, see Fig. 4-12.

Roll Construction. A typical roll for a modern troughing idler is shown in Fig. 4-8. The shell is a steel tube; the two heads are malleable-iron castings with turned edges pressed into the shell and spot-welded to it, after which the corners are spun over and rounded to avoid damage to the belt should it ride over the edge. Each head is bored to a press fit for the outer race or cup of a roller bearing and has outside the bearing a set of washers which form a "labyrinth" seal for grease. An inner steel tube ties the two heads together, acts as a reservoir for grease, and serves to conduct the lubricant from one end of the roll to the other; at the same time it prevents grease from escaping into the hollow of the roll. The roll shaft is threaded at each end for a malleable-iron nut by which the roller bearings can be adjusted for internal clearance. The nut has a flange large enough to shield

the labyrinth washers from bumps and blows, and the hexagon flats of the nut fit the slotted top of the malleable-iron supporting bracket. This preserves the adjustment of the roller bearings and allows the roll to be lifted out of the brackets when necessary without disturbing anything else. The roll shaft is drilled and tapped at one end for a fitting to take the nozzle of a grease gun. Grease under pressure enters the central tube and flows outward through the roller bearings and into the grease seals. These are not designed to prevent entirely the escape of grease, but rather to allow it to leak out gradually and

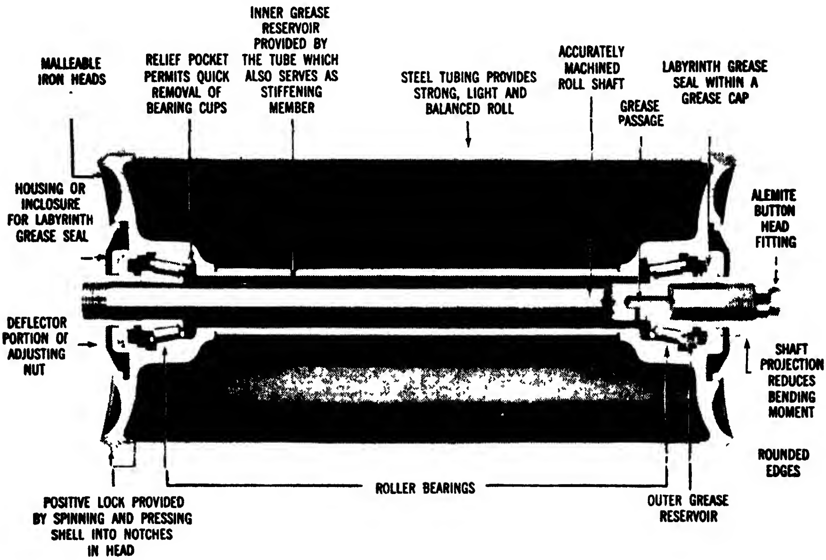


FIG. 4-8. Cross-sectional View of a Roll (Link-Belt Company)

slowly and in so doing to form on the outside of the seal a collar or ring of grease which hinders dirt or water from entering the bearings.

Fig. 4-9 shows a roll similar to that shown in Fig. 4-8; the heads are made of steel, and the central tube is large enough in diameter to take the roller bearings. The tube and the bearings are furnished as a unit by the makers of the bearings. The central shaft is provided with an adjusting nut at one end.

In the rolls shown in Figs. 4-8 and 4-9 the roller bearings are adjusted at the factory before the idlers are shipped, and, although they can be adjusted in the field to take up clearance due to wear, the amount of wear is usually slight, and adjustment may never be required. In the Robins idler roll shown in Fig. 4-10 a coil spring back

of each roller bearing automatically takes up what small amount of slack may form in the bearings. A labyrinth seal with a cork washer holds the grease in.

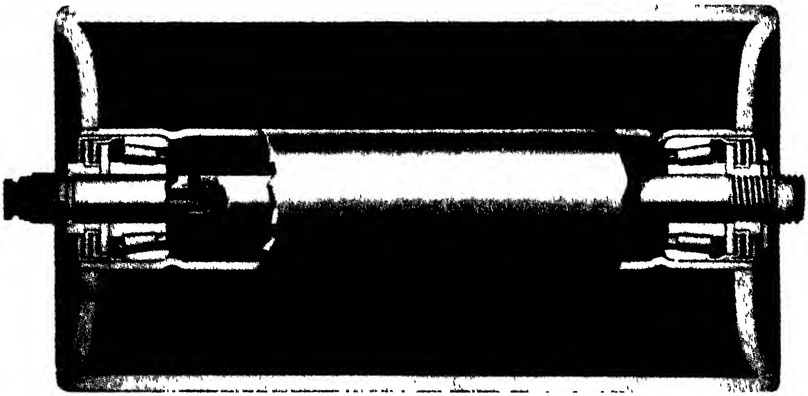


Fig. 4–9. Cross-sectional View of a Roll. (Chain Belt Company.)

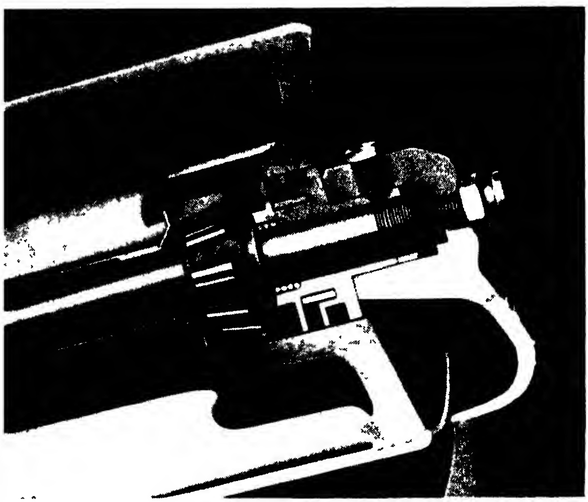


Fig. 4–10. Idler Roll with Spring Adjustment for Roller Bearing. (Robins Conveying Belt Company.)

Cast-iron rolls covered with soft rubber ¾ inch thick or solid rubber rolls are sometimes used to support the belt at loading points when the impact of falling pieces is very severe.

Fig. 4-11 shows a test made by Stephens-Adamson Manufacturing Company to prove the benefit of rubber rolls at the loading point. The upper run of the belt in the picture is coming toward us; it is supported by a horizontal roll at R the left half of which is steel, the right half of rubber. Pressing down on the belt are two weighted pulleys with lugs welded to the rims so that the impact of the lugs with the cover of the belt will imitate the action of lumps striking the belt. The pulleys are not directly over the supporting roller, but in advance of it as the end of a loading chute should be. The test showed little or no wear on the right-hand half of the belt which was

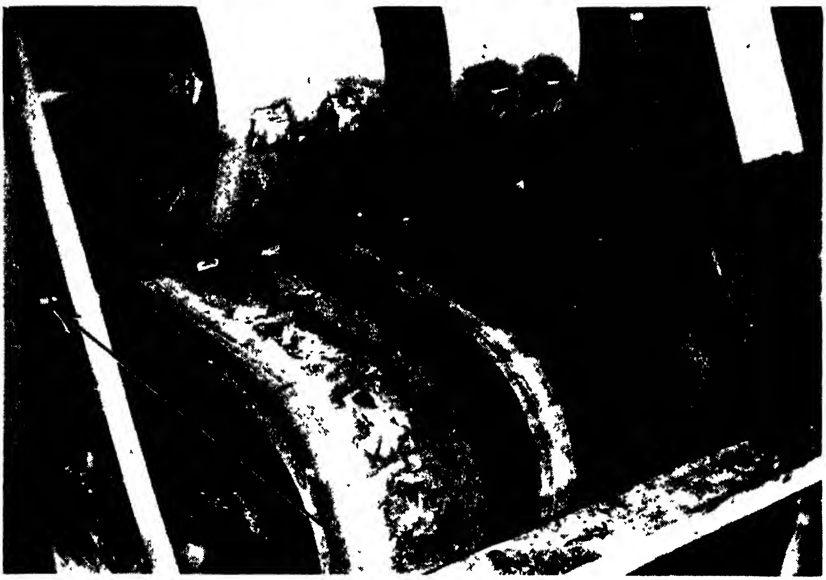


FIG. 4-11. Effect of Impact at Loading Point.

supported by the rubber roll, but serious wear on the left-hand half carried by the steel roll.

Where rolls made of steel tubing are corroded by salt air or chemicals, rolls of cast iron will often last enough longer to pay for their higher price.

Corrosion of Idler Rolls. Patches of coal dust occasionally adhere to the surface of rolls and pulleys not only from spill but also from dust which absorbs moisture from the air and adheres to a cold surface. Corrosion of the metal is likely to start under such accumulations, and it is hard to remove them. Several methods have been adopted to cure the trouble: (1) sand blasting the surfaces and coating

them with zinc; (2) "granitizing," that is, enameling them as a cooking utensil is enameled; (3) covering them with rubber.

Ball-bearing Idlers for H. C. Frick Coke Company. Fig. 4-12 shows an important form of troughing idler made by the Stephens-Adamson Manufacturing Company and in operation since April, 1924. These three-pulley idlers are used with 48-inch 8-ply belts carrying run-of-mine coal at 500 feet per minute. The troughing idler consists of three pulleys each 17¾ inches long, made of a 7-inch steel tube welded to malleable-iron heads pressed tight on a 1-inch shaft. Each end of

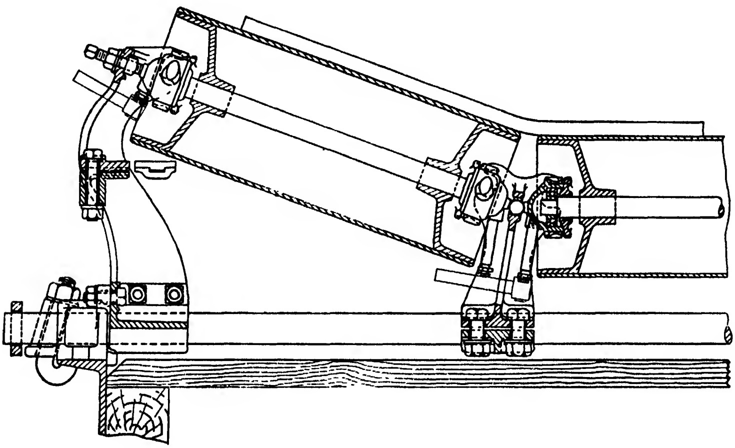


FIG. 4-12. 48-inch Three-pulley Idler for H. C. Frick Coke Company at Colonial Dock.

the shaft is journaled and fits into the inner race of an anti-friction bearing. The illustration shows a high-grade ball bearing with 5/16-inch balls. The bearings are mounted in malleable-iron shells provided with adjustment in all directions on knife-edge supports.* Each shell has its own pipe connection to receive grease from a high-pressure gun, and the leakage of grease is prevented by a cap provided with sealing grooves. The conveyor system in which these idlers are used is more than 4 miles long. To expedite lubrication there is a car running on a track alongside the conveyors; on the car are an air-compressor and a reservoir holding a barrel of grease to which a hose and the gun are attached.

The brackets that hold the bearings are mounted on a rock-shaft extending across the conveyor. On this shaft as a pivot, the whole idler can be turned down out of working position for inspection or

* Smith reissue patent, 16432 of 1926.

removal. In the working position, the idler leans 2° out of the vertical in the direction in which the belt runs.

Although the angle of troughing is only 20°, the belts regularly carry 50 per cent more than the standard ratings (see page 194) and there is no spill; they run straight, side-guide idlers are not used, and the consumption of power is exceedingly low. The return idlers are steel tubes, 7 inches in diameter and 57 inches long, mounted on 1¼-inch shafts journaled at the ends to take anti-friction bearings like those used in the troughing idlers.

After three years of experience with the belt conveyors at Colonial Dock the H. C. Frick Coke Company built another belt conveyor

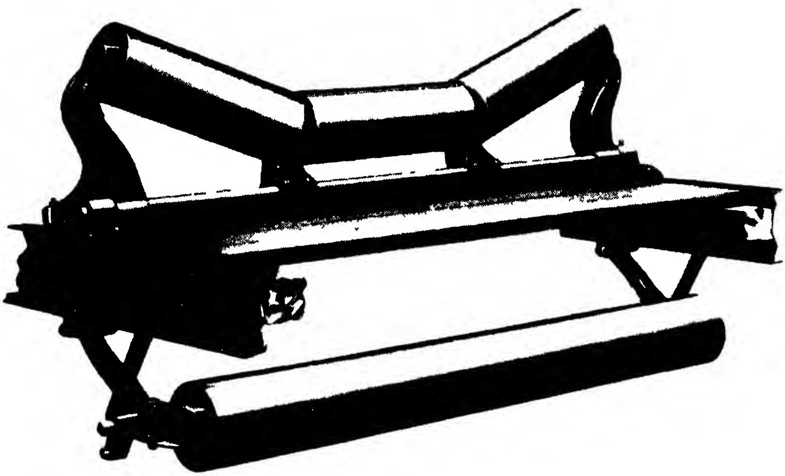


FIG. 4-13. Three-roll Idler for 60-inch Belt.

system for carrying the output of several coal mines to the Monongahela River at Palmer Dock. This has six 48-inch belts and five 60-inch belts running at 545 feet per minute. The idlers are like those at Colonial Dock except that the rolls have 8-inch-diameter shells instead of 7-inch. Fig. 4-13 shows the assembly of the troughing idler and the return idler. The latter has a lever suspension which permits the roll to be dropped down for examination and repair and then returned quickly to its operating position.

The Sacon idler of the Stephens-Adamson Company is of the live-shaft type and is similar to the Frick idler, but without the tilting arrangement. The malleable-iron brackets are bolted to an inverted

angle. The rolls are 6 or 7 inches in diameter and have double Timken roller bearings mounted in housings supported on trunnions, Fig. 4-13A.

Idlers for Grain Conveyor. Grain is a free-flowing material and is usually delivered to a belt through a closed spout. It flows rapidly, and when it meets the surface of the belt it spreads out and forms a shallow pile. For this reason grain belts are generally wider than belts for other materials, and since grain weighs no more than 50 pounds per cubic foot it is not necessary to place idlers so close to-

FIG. 4–13A. Housing and Double Bearings for Live-roll Shaft.

gether. Five feet is an ordinary spacing, and if the belt receives its load at one point only, concentrators every 10 or 15 feet will keep the grain on the belt. If the belt is loaded at many points along its length, the idlers may be spaced 4 feet apart with concentrator rolls on every stand. Fig. 4-14 shows a "transfer" belt in a grain elevator receiving grain from one of a row of spouts under scale hoppers. Each stand has concentrator rolls to prevent the grain from scattering as it leaves the spouts, and the turned-up edges of the belt limit the spread of the grain sideways. The idler spacing is 4 feet.

Grain conveyor idlers of the style shown on page 11 are still in use, but many have been replaced by the kind shown in Table 4-C. These are also preferred for new installations; the continuous roll will not hurt the belt, as may happen with spaced pulleys of narrow face,

maintenance is easier, and the better bearings help to reduce the fire hazard.

Idlers For Picking Tables. A picking table is a slow-moving belt conveyor that carries coal or ore in a thin layer so that the men who stand beside the belt can see the separate pieces, turn them over, pick out the bad, and let the good pass on. The rolls are 5 or 6 inches in diameter, similar in construction to standard rolls. The inclined rolls

Fig. 4–14. Grain Conveyor with Numerous Loading Spouts.

are short and make only a shallow trough. Table 4-D shows the usual construction.

Flat Roll Idlers used for picking tables, sorting conveyors, and conveyors with plow discharge have rolls 5 or 6 inches in diameter. For package conveyors or light assembly conveyors, the rolls are usually smaller and are fitted with ball bearings.

Grease Lubrication. The primary purpose of grease lubrication is to keep wet and dirt out of the bearings so that the polished surfaces of balls, rollers, and races stay bright and clean. If they become rusty or pitted, the coefficient of friction goes up, and the parts wear out.

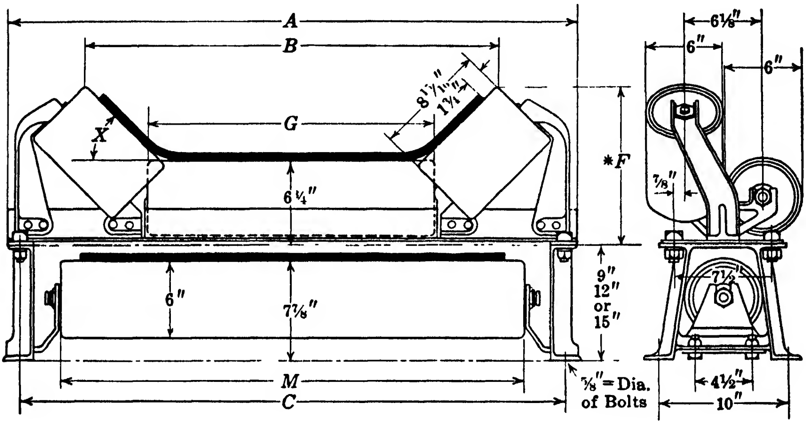
The grease fittings on the idler should be wiped off before the gun is attached, or else dirt will be forced in with the grease.

Care should be taken not to force too much grease at high pressure into the rolls; the grease seals may be pushed out of place, or what

TABLE 4-C

IDLERS FOR GRAIN CONVEYORS

Link-Belt Company



Width of Belt	A	B		C	G	M
		$X=35°$	$X=45°$			
24	35	$24\frac{9}{16}$	$22\frac{1}{2}$	33	$12\frac{7}{8}$	$26\frac{7}{8}$
30	41	$30\frac{9}{16}$	$28\frac{1}{2}$	39	$18\frac{7}{8}$	$32\frac{7}{8}$
36	47	$36\frac{9}{16}$	$34\frac{1}{2}$	45	$24\frac{7}{8}$	$38\frac{7}{8}$
42	53	$42\frac{9}{16}$	$40\frac{1}{2}$	51	$30\frac{7}{8}$	$44\frac{7}{8}$
48	59	$48\frac{9}{16}$	$46\frac{1}{2}$	57	$36\frac{7}{8}$	$50\frac{7}{8}$

leaks past them may be thrown off the roll by centrifugal force and create a nuisance.

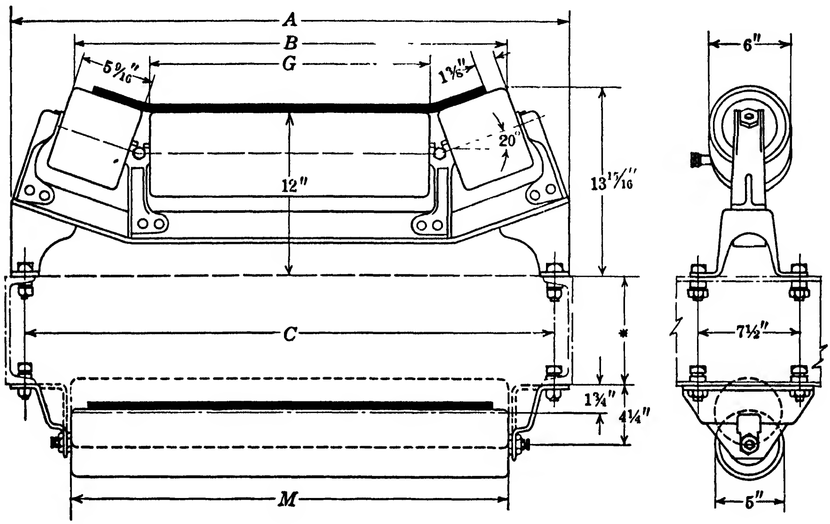
Idlers with Prelubricated Bearings. Several manufacturers of belt-conveyor equipment furnish idler rolls with ball bearings the races of which are enclosed in sheet metal shells with a grease seal. One method of using them is shown in Fig. 4-6, Brouwer patent 2,073,957 of 1937. The rolls have no through shafts, the heads of the rolls take the outer races of the ball bearings, and the cup-shaped inner races

are a tight spring fit over the ball-shaped ends of a short stub shaft which is clamped to the supporting stand. The bearings are filled with grease at the factory; since they are closed on one side and have

TABLE 4-D

IDLERS FOR PICKING BELTS

Link-Belt Company



Width of Belt	A	B	C	G	M
24	35	26	33	15	$26\frac{7}{8}$
30	41	32	39	21	$32\frac{7}{8}$
36	47	38	45	27	$38\frac{7}{8}$
42	53	44	51	33	$44\frac{7}{8}$
48	59	50	57	39	$50\frac{7}{8}$
54	65	56	63	45	$56\frac{7}{8}$
60	71	62	69	51	$62\frac{7}{8}$

Depth of channel should be sufficient to suit span.

a grease seal on the other, lubrication is said to be required only at long intervals. There is no grease-gun attachment.

Some greases lose their lubricating quality in time by a separation of the liquid part which is oil from the solid part which is a soap. Other greases are better in this respect and under good conditions last

long. Dead-shaft rolls of conventional form under observation in practical use have been known to run more than ten years on the grease put into them at the factory, with no new grease at all. These rolls had good grease seals. It is probable that prelubricated rolls of this kind will come into wider use in the future.

Fig. 4-15 shows a roll taken apart after more than ten years' hard work on a coal conveyor; the bearings were in good condition, and the original grease hardly discolored. The grease openings had been welded shut when the test started, so that no new grease could be added.

What Makes a Good Idler Roll? The following are characteristics of a good idler roll: (1) means of keeping dirt and wet away from the

FIG. 4-15. Idler Roll after Ten Years' Run on the Original Charge of Grease.

bearings; (2) ease and certainty of lubrication; (3) accurate balance of the roll around its axis of rotation; (4) accurate and permanent alignment of the bearings; (5) thickness of shell enough to resist corrosion and the effect of bumps and blows; (6) construction permitting access to the bearings without dismantling the entire idler.

Return Idlers. These were formerly a series of spaced pulleys set-screwed to a shaft which ran in plain bearings hung under the conveyor stringers. The edges of the separate pulleys were likely to cut the belt when it had any side movement, and if there was any settling of the conveyor supports, the shaft would not turn freely. To avoid damage to the belt, the rolls are now made of steel tubing some inches longer than the belt so as to allow the belt to move sideways without hitting

the conveyor frame or the bearing supports. For most work the shafts do not revolve but are fastened to brackets at each end, while the roll with ball or roller bearings pressed into the heads of the roll runs loose on the shaft. In the style shown in Table 4-A the shaft has a grease fitting at each end and is milled to fit a slot in the hanger, so that the shaft and roll can be lifted out. For belts over 48 inches in width, the return roll in some designs is tight on its shaft and the shaft runs in roller bearings in ball and socket hangers mounted in the conveyor stringers. With this "live-shaft" construction it is necessary to use tie rods or cross members to keep the conveyor stringers at the proper distance apart. In the dead-shaft construction shown in Table 4-A the milled ends of the shaft keep the shaft hangers properly spaced

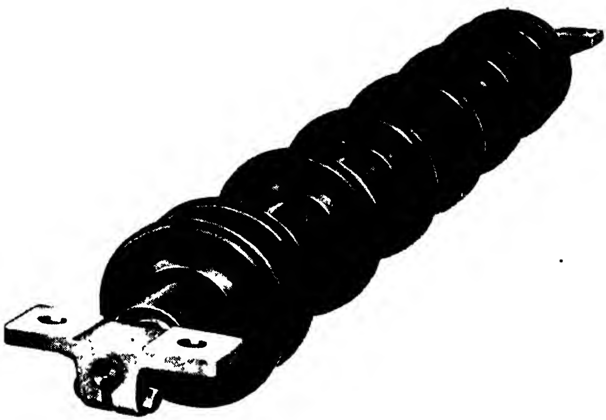


FIG. 4-16. Rubber Disc Return Idler. (Robins Conveying Belt Company.)

and, so long as the shaft does not bend, the bearings on the roll are in correct alignment.

Rubber Disc Idlers (United States patent 2,052,900) as made by the Robins Conveying Belt Company, Fig. 4-16, consist of a series of $5\frac{1}{2}$-inch-diameter discs, five in number for a 16-inch belt, to twelve for a 60-inch belt, fastened to a central steel tube. In the ends of the tube are mounted ball bearings which run on ball-end stub shafts, like those in Fig. 4-6. Advantages claimed are light weight, no accumulation of dirt, and less wear on the belt.

Why Belts Run Crooked. A conveyor belt run flat over cylindrical rolls will run straight if the rolls are set square with the center line of the belt, and if the belt itself is made straight in the factory and properly spliced in the field. The evidence on this point is that side-

guide idlers are seldom required on the return run of belt conveyors. However, if the belt is run over troughing idlers of any kind, there comes a tendency to run crooked; this occurred with the spool idlers used on old grain conveyors, and with "dishpan" idlers also, because, if the belt shifted to one side by reason of eccentric loading or bad alignment, it acted as a belt does in running on to a crown-face pulley, that is, it moved toward the large part of the pulley.

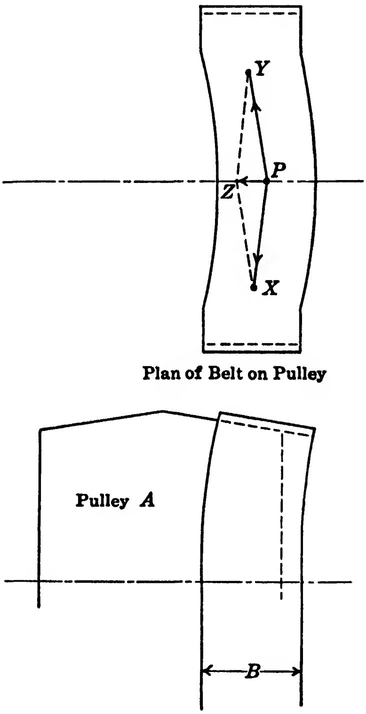


FIG. 4-17. Belt on Crowned Face Pulley.

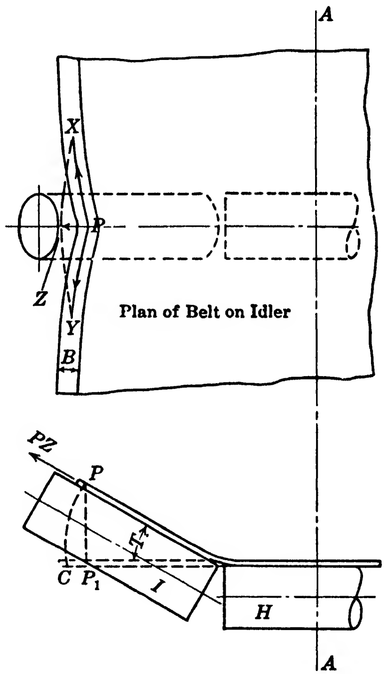


FIG. 4-18. Steering Effect of a Troughed Roll Idler.

Fig. 4-17 shows why a belt centers itself on a crown-face pulley. If the belt B is placed on the driven pulley A to one side of the center of the crown, it lies on a conical surface. The flexible belt assumes the position shown, the pulls in the belt are in the lines PX and PY, and the resultant force PZ pushes the belt higher on the crown. When the belt is centered on the crown, the forces tending to shift the belt are balanced and the belt runs central on the pulley. The action of a conveyor belt on an inclined troughing roll is similar.

In Fig. 4-18, AA is the center line of a conveyor belt troughed over

the horizontal roll H and two inclined rolls of which I is one. When the belt is troughed the edge is lifted and point C passes to point P. Consider a strip of belt of width B. The inclined roll forces this strip from a straight line as shown in the plan view. The pull in the strip is along lines PX and PY, resulting in a force PZ which tends to push the belt higher on the inclined roll. If the troughing roll at the opposite end of the idler acts with an equal amount of deflecting force, the two forces neutralize each other and the belt runs straight; if the deflecting force in the opposite roll is greater the belt will tend to move to that side.

If the roll is hard to turn, the forces PX and PY are greater and PZ is greater than if the friction is low. This is one reason why idlers with anti-friction bearings are not so likely to cause crooked running as the old-style idlers with plain bearings. The angle of troughing also has its influence. If T, the angle between the two positions of the edge of the belt, is reduced, PZ is reduced. If the roll is not inclined at all or if it were possible to prevent the belt from flattening between idlers, PZ would be zero. The distance CP_1 represents the distance by which the edge of the belt is forced from a straight line by troughing, and it is this displacement combined with the tension in the belt that produces the force PZ. For an idler with $T = 20°$ the deflecting force is less than one-fourth as much as for the original 45° idler shown in Fig. 2-15.

If the troughed belt is under tension and the idlers are set close together, the edge of the belt in the sag between idlers is never so low as point C but at a point higher up toward P. The effect is the same as reducing the angle T; hence the belt will run straighter if it is not allowed to run slack. Too much sag between idlers shows a condition in which a belt tends to run crooked. This explains why troughing idlers should be spaced closer together near the foot of a conveyor where the belt tension is least than toward the head where the tension is greatest.

If the force PZ in the roll I is not balanced by the corresponding force in the roll opposite to it, the belt would tend to shift sideways on the horizontal roll H; the difference of the two forces is resisted by the friction of the belt on the surfaces of all three rolls. If the belt is loaded, the resistance is greater than when the belt is empty; hence a load put on a belt helps it to run straight. If the horizontal roll is relatively long in proportion to the width of the belt, it carries more of the load than comes on the other rolls and the belt is less likely to be shifted out of line. When a belt carries an unsymmetrical load, the troughing rolls on one side of the conveyor carry more weight than

those on the opposite side. The roll friction on the heavily loaded side is then greater, and the sum of the outward resultants on that side might be expected to shift the belt outward. But if the load is deeper on that side, gravity will resist the slight tendency to lift the belt and its load, and the load will sink to a low position. Under all circumstances, a belt that is centrally loaded will tend to run straighter than a belt loaded off to one side.

A belt with too many plies or too stiff to trough properly gets little or no guiding effect from the horizontal roll and hence is more likely to run out of line. The same is true of narrow troughed belts; they tend to run crooked.

Belts Running Crooked: Causes and Cure. The fault may be in the belt, in the idlers, in the manner of loading, in the drive, or in the supports. It may be caused by wind blowing against one side of the conveyor.

1. If one place on the belt persistently runs to one side, the belt itself may not have been made straight at the factory. More likely the fault is in the splice. The splice must be square with the center line of the belt; the best way to square the cut is to mark a center line equidistant from the edges of the belt for a distance of 5 feet or more from the splice and work from that line, not from the edges of the belt. If the fasteners at a splice must be renewed or replaced, square the ends at the splice as if for a new belt.

2. A belt may have run straight for a time, then start to run crooked. Examine the belt; if one edge is worn thin, that edge may be stretching. Sometimes a worn edge absorbs moisture and shrinks. The remedy is to cut out the worn belt and splice in a new length, or else put on a new belt.

3. If the belt is too stiff or too thick to trough properly the trouble may sometimes be cured by running the belt slower so that it carries a heavier cross section of load. But this will not help an empty belt, and the remedy then is to use enough swiveling or self-aligning idlers to steer the belt straight.

4. When a belt throughout its length runs to one side at a certain idler, the fault is not in the belt but in that idler or in one behind it. The idler may not be level, it may be out of square with the belt, or a roll may be hard to turn from no lubrication or from a ball or roller bearing which is broken or out of adjustment. The forces which deflect a belt are sometimes cumulative and may be corrected by adjusting one or two idlers behind the one where the deflection is greatest. When the idlers finally have been set so that the belt runs true, their positions should be marked so that after repairs the idler stands can be put back

in the proper places. The bases have slotted holes to allow some adjustment.

5. Old-style idlers with steep troughing angles will cause crooked running. The old remedy was side-guide idlers (page 109). A better one is to use self-aligning idlers (page 112).

6. If the crooked running is worst at the foot of the conveyor or behind the loading point where the belt tension is low, put in more troughing idlers to support the belt at closer intervals. As a temporary expedient, use side-guide idlers until the fault is corrected, then remove them.

7. Belts fed from the side of the conveyor often run crooked because the load is not central on the belt. The remedy is to correct the chute if possible or else use self-aligning idlers. If a lopsided load due to rushes of material, as from a crusher, causes the belt to run off side, a feeder should be used to stop the rushes, or as an expedient, baffle bars can be hung in the chute to check the flow of material.

8. If a belt crowds to one side near the head pulley, the head shaft may not be square with the center line of the conveyor. It is important to establish the center line of the conveyor accurately and mark it permanently, before the machinery supports and conveyor stringers are set. A piano wire stretched between terminals will serve to establish the center line in most conveyors. Sighting with a surveyor's instrument is the best way for long conveyors.

9. Any of the faults mentioned here can be masked and the belt forced to run straight if weighted take-ups are overloaded or if screw take-ups are set to pull the belt out of line or put an excessive tension on it. This is bad practice, for it may harm the belt or weaken the splice. Head shaft, foot shaft, and all other shafts should be set square with the belt and kept square.

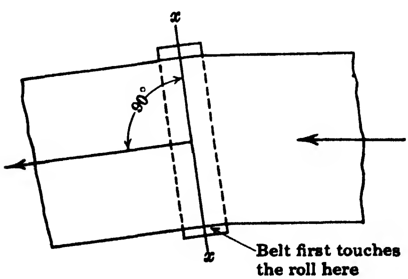


FIG. 4-19. Steering Effect of a Flat Roll Idler.

10. If the belt runs crooked on the return run:

(*a*) See that the idler shafts are square with the run of the belt.

(*b*) If the belt chafes or interferes at one spot the conveyor supports may be out of line. The fault can be corrected to a certain extent by setting a few roll shafts out of square with the belt. The belt will shift to the side where it first touches the roll, Fig. 4-19.

(*c*) See that the rims of the rolls are clean and have no accumula-

tion of dirt on them. Crusts of dust mixed with atmospheric moisture sometimes build up on one end of a roll and make the belt run offside.

(*d*) See that the set screws in the rolls are tight. If a roll shifts endways, it may cut the belt or allow it to run off the roll and drag on the roll shaft.

(*e*) Side-guide return idlers, see page 103, should not be used to control the path of the belt. They will wear the edge off and ruin the belt.

(*f*) For the Robins "return trainer" see page 114.

Devices to Make Belts Run Straight. The old original device to keep belts in place is the side-guide idler bearing against the edge of the belt. With 45° troughing, see page 14, they were indispensable, but a belt properly installed on 20° idlers should not need them. On

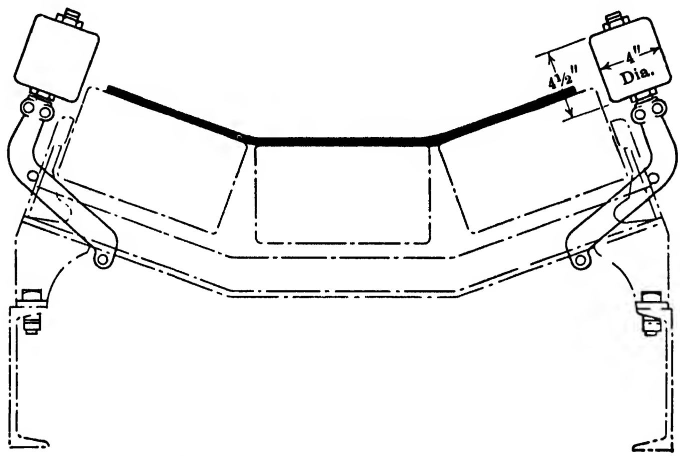


FIG. 4-20. Side Guide Idler Clamped to Carrier Stand.

wide belts where a good proportion of the belt and its load rests on the horizontal roll they are seldom required. In any event they do not get at the cause of the trouble, and in trying to force a belt into place they may ruin it. They should be regarded as temporary expedients only. When they are used they should be set close to a troughing idler so that they act on the belt where it is supported; if they are put midway between idlers, the sag of the belt may cause the edge to miss the face of the side-guide pulley, get under it, and then chafe against the stand. Uncertainty of this kind is avoided altogether by a form of idler which clamps on the idler stand as shown in Fig. 4-20 (Link-Belt Company).

The effect of side-guide idlers is generally bad. They wear the

belt at its most vulnerable place and open the way for dirt and wet to get between the plies of fabric. If the belt is badly out of line, the pressure against the idler may be enough to bend or fold the belt for an inch or two all along the edge so that a crack develops there and splits the belt.

Steering the Belt by Skewing the Troughing Rolls. One way is shown in the Mason patent of 1907 (Fig. 4-21). It is to set the inclined rolls with a rake forward in the direction of belt travel. Each inclined roll then acts on the part of the belt in contact with it to steer it toward the center of the conveyor just as the skewed roll shown in Fig. 4-19 shifts the belt toward the side where the belt surface first touches the roll. If the belt is running central on the idlers, nothing happens, but if it for any reason runs to one side, more of the belt surface on that side is acted on by the skewed roll and the belt is guided back to a

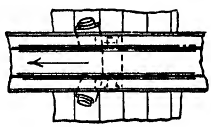


FIG. 4-21. Guiding a Belt by Skewing the Inclined Pulleys.

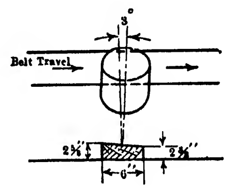


FIG. 4-22. Guiding a Belt by Tilting the Idler Stands and Pulleys.

central position. This device was not used more than a few times; a disadvantage is that the rubbing on the skewed pulley tends to wear away the rubber on the pulley side of the belt.

Another way to skew the troughing rolls is to tilt the whole idler forward in the direction of belt travel, Fig. 4-22. It is done by beveling the plank which carries the idler or, if the idler is on a steel frame, by putting washers under the legs of the frame. If in the 36-inch size shown in Table 4-A the height to the low end of the troughing roll is about $10\frac{1}{4}$ inches and to the high end $14\frac{5}{8}$ inches, then a tilt of 3° forward in the direction of belt travel will set the low end 10.25 sine 3° = 0.52 inch forward and the high end 14.62 sine 3° = 0.76 inch forward; that is, the troughing roll will be about $\frac{1}{4}$ inch out of square with the travel of the belt. As compared with the Mason device this small amount is usually not enough to hurt the belt, yet it exerts a guiding influence which is often effective enough to make a belt run straight. The two methods here described are good only if the belt runs in one direction.

When idlers are shipped on beveled planks it is customary to mark an arrow on them at the factory to show the direction in which the

belt should run. One case is known where a foreman "reasoned it out" that the arrows were wrong, installed a number of conveyors with the idlers the other way about, then when the belts tried to run off the idlers, he put planks all along the sides of the conveyor to keep the belts in place.

In the three-pulley 20° troughing idler shown in Fig. 4-12 the forward tilt of the pulleys is only 2°, but the 48-inch belts used on them run perfectly straight without the use of side-guide idlers. The belts have a good contact with the wide face of the middle pulley of the idler, and since the inclined pulleys are set at the low angle of 20° the forces which tend to deflect the belt are comparatively small.

Automatic Belt Steering. When a belt carries heavy material like ore and is loaded unsymmetrically, i.e., more to one side of the center

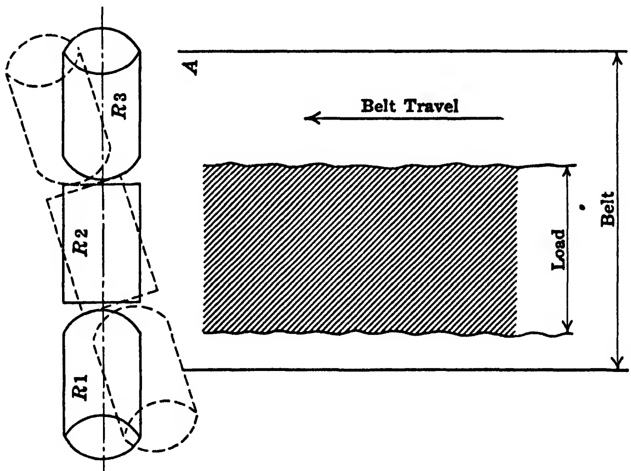


FIG. 4-23. Belt with Load off Center.

than the other, the load will seek the low point and the edge of the belt on the side least loaded will ride up on the troughing rolls to an extent which cannot be controlled by tilting the idlers forward and which it is dangerous to try to stop by side-guide idlers bearing on the edge of the belt. Often the conditions of loading or the wetness or dryness of the material are such that the load cannot be centered on the belt at all times. This may happen when belts are fed from the side. Lump or dry material may come to rest on the far side of the belt; damp or fine material will stay on the near side. Users of belts, particularly in the western smelters, have therefore tried various automatic steering devices, most of which are troughing idlers in modi-

fied form. Fig. 4-23 shows a belt with a lopsided load supported on a troughing idler with rolls $R1$, $R2$, $R3$. The edge of the belt at A is in a dangerous position; the edge of $R3$ may cut it. An obvious correction is to turn the idler stand so that the rolls take the positions shown dotted. This can be done by: (1) mounting the idler stand on a central pivot pin or a turntable; (2) providing means attached to the stand and operated by the belt, to turn the rolls toward the dotted position before the edge of the belt reaches the dangerous position at A; (3) providing similar means to prevent the belt from going too far in the other direction and then restoring it to its central, or at least to its neutral, safe position.

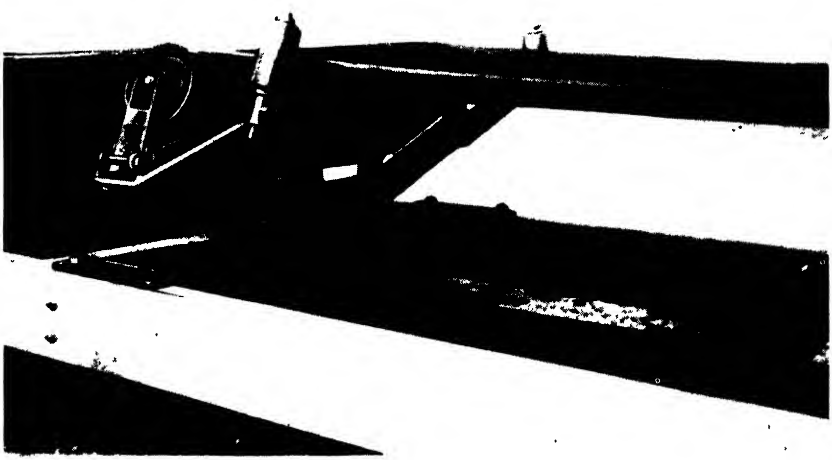


FIG. 4-24. Self-aligning Idler. (Link-Belt Company.)

Swiveling troughing idlers which act in the way described are shown in patents granted to Sibley, 1916; Forbes, 1924; Nelson, 1924 and 1926; Cuddihy, 1929; Robins, 1934; Sayers, 1935; and others. Most of the earlier designs were crude and clumsy and did not come into general use. The increasing use of belt conveyors has stimulated manufacturers into building these "self-aligning" idlers with lighter and better steel frames, sheet-steel rolls with ball or roller bearings instead of cast-iron rolls with plain bearings, and with better lubrication and better protection against wet and dirt. These modern idlers steer the belt easily and positively and make it unnecessary to use side-guide idlers or tilt the stands of the troughing idlers.

A modern self-aligning idler is shown in Fig. 4-24. A set of

standard troughing rolls is mounted on a frame with a central pivot carried by a cross member resting on the conveyor stringers. Two arms projecting from the idler frame carry "actuating rolls" which normally stand clear of the belt. If the belt gradually moves out of center and up on one of the troughing rolls (as *R*3, for instance, in Fig. 4-23) its greater contact on that side causes the frame to swing slightly toward the dotted position and steer the belt back to its central position perhaps without touching the "actuating rolls," but if the belt should move suddenly to one side it will push against one of the actuating rolls and quickly swing the frame and the rolls into the dotted position. The belt is then steered toward the other side, its

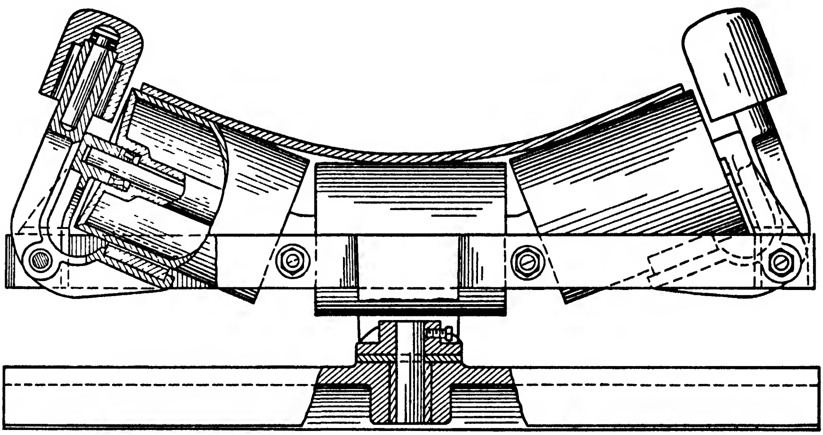


FIG. 4-25. Self-aligning Idler with Brakes.

opposite edge touches the other actuating roll, and the idler and the belt assume a central position again. In one make of self-aligning idlers, the actuating roll is mounted on a hinged arm backed up by a spring so that, if a large lump should project beyond the edge of the belt and hit the actuating roll, the roll and its supporting arm will yield in the direction of belt travel and not be damaged. The positive action of idlers having actuating rolls depends on having the rollers on the side of the idler toward which the belt comes, hence they cannot be used on a belt that is reversible in direction of travel.

The S. D. Robins patent 1,963,099 of 1934, Fig. 4-25, shows a method of steering a belt which is independent of the direction of belt travel. The three troughing rolls are mounted on a frame that is pivoted at the center and carries at each end an actuating roll mounted on a bell-crank lever. A brake shoe on the lower arm of the lever normally clears the rim of the troughing roll. If the belt shifts side-

ways enough to touch the actuating roll, the brake is applied to one roll with the result that the drag of the belt on its rim causes the idler frame to swing to a skewed position and steer the belt back from the offside position. A drawback to the construction shown in the drawing of the patent is that the first movement of the idler toward the skewed position jams the actuating roll harder against the edge of the belt.

Another device that is operative on a reversible belt is shown in the Nelson patent 1,572,555 of 1926. An idler mounted on a pivot carries beyond each troughing roll a disc loose on the roll shaft and independent of the roll. If the belt shifts too far to one side it runs onto the edge of the disc: the disc is weighted on one half, and because of the eccentric weight it resists turning sufficiently to put a drag on the edge of the belt. When the idler has swung to the skewed position, the belt is steered back to the central position and contact with the weighted disc ceases. In service, the rotation of the disc with its unbalanced weight is likely to cause wear on the bore of the disc or its

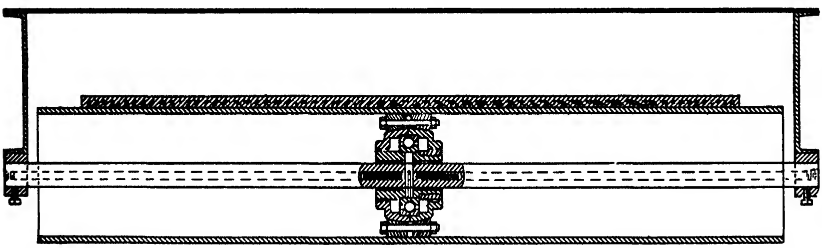


Fig. 4-26 Flat Roll Training Idler. (Robins Conveying Belt Company.)

bushing, and the device becomes noisy. It is shown in the catalog of the Link-Belt Company.

Those who have had much to do with belt conveyors know that belts will sometimes run crooked for reasons hard to discover, will change from day to day, and misbehave in ways hard to cure. They find that the use of self-aligning idlers corrects much trouble of that kind and does it in a way which does not hurt the belt.

Self-aligning Return Rolls. The S. D. Robins patent 1,833,180 of 1931 (Fig. 4-26) shows a means to steer a return belt. The roll shaft is fixed and carries at midlength a hub pinned to the shaft. By adjustment of the roll shaft the pin is set in a position somewhat off the vertical, its top end leaning forward. A ball bearing for the roll is mounted on the central hub. When the belt runs central, it is balanced on the central bearing; but if it runs to one side, the greater drag of the belt on that side skews the roll forward with the oblique pin as a pivot.

This skewing guides the belt toward the opposite side, but as that end of the roll has risen slightly by reason of the oblique position of the pin, too great or too rapid shifting of the belt is checked and the belt is restored to its central position gradually.

Spool or flared idlers have nearly passed out of use. Page 25 states their merits and demerits. Modern troughing idlers with anti-friction bearings and improved lubrication are so much better than old-style troughing idlers that little or nothing is to be gained by the use of spool idlers for general conveying work. For special purposes, they are still made by several manufacturers in the form shown in Fig. 4-27. A steel tube is fastened to two malleable-iron bell-shaped ends and mounted on a straight shaft supported in trunnion bearings.

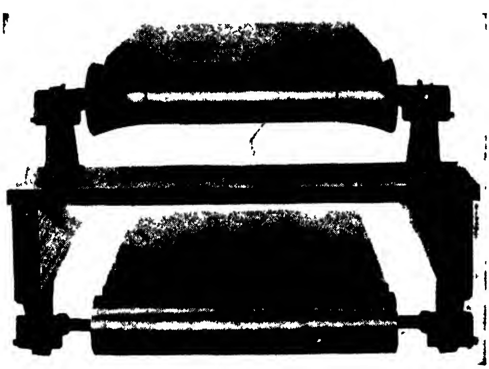


FIG. 4-27. Uniroll Flared Idler with Oil Lubrication. (Link-Belt Company.)

On wagon-loaders, graders, and similar portable machines that use belt conveyors, it is often important to reduce weight and save space. Spool idlers of small diameter and deep concavity are sometimes used in such cases. The wear they cause on the under side of the belt is not important, because the life of the belt in such machines is generally determined by other factors, such as wear on the carrying side or on the edges, or by accidents.

Table 4-E gives weights of one manufacturer's list of troughing idlers, including cross members, stands (if any), brackets, pulleys, and rolls. These are useful for estimating shipping weights and loads on structural supports. The lines "moving parts only" give weights of rolls and the rotating parts of the roller bearings. These enter into calculation of the horsepower for the conveyor. The table also gives weights of return idlers including rolls, shafts, and end brackets, and

TABLE 4-E

WEIGHTS OF COMPLETE IDLERS AND WEIGHTS OF MOVING PARTS

Width of Belt, inches	14	16	18	20	24	30	36	42	48	54	60
						Weights in Pounds					
Troughing, Type 90, 4-in. steel rolls,											
⅝-in. shafts, ball bearings	30	32	34	36	40						
Moving parts only	14	15	16	17	19						
Return, Type 91, 4-in. steel rolls	17	18	20	21	24						
Moving parts only	12	13	14	15	17						
Troughing, Type 70, 5-in. steel rolls,											
¾-in. shafts, roller bearings	44	47	50	54	59	76	87	95	104		
Moving parts only	26	27	29	31	34	39	44	49	54		
Return, Type 71, 5-in. steel rolls	25	27	29	31	34	42	48	54	60		
Moving parts only	18	20	21	23	26	31	36	41	46		
Troughing, Type 80, 6-in. steel rolls,											
¾-in. shafts, roller bearings	52	55	59	63	71	96	106	120	132		
Moving parts only	35	37	39	41	45	52	58	65	71		
Troughing, Type 80, 6-in. cast-iron rolls,											
¾-in. shafts, roller bearings	70	75	81	87	99	130	148	167	184		
Moving parts only	53	57	61	65	73	86	100	112	123		
Return, Type 81, 6-in. steel rolls						same as Type 41					
Moving parts only						same as Type 41					
Troughing, Type 40, 6-in. steel rolls,											
¾-in. shafts, roller bearings	57	60	64	68	74	98	108	122	134	145	156
Moving parts only	35	37	39	41	45	52	58	65	71	78	84
Troughing, Type 40, 6-in. cast-iron rolls,											
¾-in. shafts roller bearings	75	80	86	92	102	132	150	169	186	203	220
Moving parts only	53	57	61	65	73	86	100	112	123	136	148
Return, Type 41, 6-in. steel rolls	30	33	35	38	42	52	59	65	71	77	84
Moving parts only	24	26	28	30	34	40	46	52	58	65	71
Troughing, Type 59, 6-in. heavy steel rolls,											
1¼-in. shafts, roller bearings			..		.	.	215	234	253	272	291
Moving parts only	.	.		..	...	..	105	114	123	132	141
Return, Type 60, 6-in. heavy steel rolls	...	...	.	..	...	.....	114	125	136	147	158
Moving parts only	..	...		..	..		77	86	95	104	113
Weights of Self-Aligning Troughing Idlers											
Type 70, actuating roll style	83	85	89	93	101	128	140	153	165		
" 80, " " "	87	90	95	101	112	140	155	169	184		
" 40, " " "	90	94	99	104	115	143	158	172	187		
" 70, counterweight disc style	96	99	102	105	114	134	146	158	170		
" 40-80 " " "	111	112	117	122	131	159	173	189	206		

of the rotating parts of the idler. The weights of idlers and of moving parts of other makes do not differ much from the figures in the tables.

A number of points must be considered in the choice of troughing idlers. If the belt is narrow, the load light, the speed low, the service only an hour or two a day, or the job one to last a short time only, an idler like Type 90 with 4-inch rolls may be good enough. For longer life, less drag on the belt, an idler like Type 70 with a 5-inch roll will be advisable; for heavier loads, longer service and still less friction loss, Type 80 with 6-inch rolls will be better; and if the conveyor is heavily loaded, or carries large lumps or is likely to get out of line, an idler with short end-brackets, Type 40, may be preferable to one with high end-brackets like Type 80. Steel rolls are standard, but if the conveyor handles salt, coke, or abrasives, or is exposed to salt air or acid fumes, a cast-iron roll will sometimes last longer. Where some of the fine coal carried back by the return belt sticks tight to the rims of the return rolls, corrosion may start under the accumulations and destroy the rims of the rolls. Zinc plating or enameling the rolls has been found helpful.

For the beneficial effect of rubber rolls at the loading point see page 96.

Spacing of Belt Idlers. The spacing of troughing idlers depends upon the weight of belt and load; the heavier the load the closer the supports to prevent the belt from sagging too much. Excessive sag tends to make the belt run crooked (see page 106); it also causes internal wear in the belt from stretch of the friction rubber and wear on its face from the slip of material as the belt passes over each idler. Likewise it adds to the power required to drive the conveyor (see page 131).

Table 4-F based on current practice gives the idler spacing considered proper for various weights and sizes of material and in addition suggests some discrimination in the choice of the idlers themselves.

Graduated Idler Spacing. Formula $H = \frac{S^2 w}{8T}$, based on the properties of the parabola, gives the sag of a suspended belt or cable, where H is the sag in feet, w the weight of the loaded belt in pounds per foot, S the span in feet, and T the pounds tension in the belt. If, for example, we take a belt with idler spacing $S = 4$ feet, w the weight of belt and load 80 pounds per foot, and T the belt tension at the foot of a conveyor 1000 pounds, then $H = \frac{16 \times 80}{8 \times 1000}$, or about 0.16 foot, nearly 2 inches. For the same belt at the head of the same conveyor

where $T = 4000$ pounds, the formula gives about ½ inch, that is, one-fourth as much as at the foot of the conveyor. Since the belt is troughed, it resists deflection and the deflection is not so much as the

TABLE 4-F

BELT WIDTHS AND IDLER SPACING

Width of Belt, inches	Type of Troughing Idler	Spacing of Troughing Idlers, inches						Spacing of Return Idlers	
		35-lb. material	50-lb. material	75-lb. material	100-lb. material	125-lb. material	150-lb. material	Type 41	Type 71
14	40	66	66	60	60	54	54	All spacings 10 ft (widths 14 to 36)	
	80	66	66	60	60	54	54		
	70	66	66	60	60	54	54		
16	40	66	66	60	60	54	54		
	80	66	66	60	60	54	54		
	70	66	66	60	60	54	54		
18	40	66	66	60	60	54	54		
	80	66	66	60	60	54	54		
	70	66	66	60	60	54	54		
20	40	66	66	60	60	54	54		
	80	66	66	60	60	54	54		
	70	66	66	60	60	54	54		
24	40	66	60	54	54	48	48		
	80	66	60	54	54	48	48		
	70	66	60	54	54	48	48		
30	40	60	54	48	48	42	42		
	80	60	54	48	48	42			
	70	60	54	48	48				
36	40	60	54	48	48	42	42		
	80	60	54	48	48				
	70	60	54	48					
42	40	54	54	48	48	42	42	9 ft	
	80	54	54	48	48				
	70	54	54	48					
48	40	54	48	42	42	36		9	
	80	54	48	42					8
	70	54	48						8
	59	54	48	48	48	48	48	10	
54	40	48	48	42				9	
	59	48	48	48	48	48	48	10	
60	40	48	48					9	
	59	54	54	54	48	48	48	10	

Note 1. For relation between size of lumps and width of belt, see page 216.
Note 2. For weights of materials, see pages 415 to 419.

formula shows, but for 20° troughing the ratio between the sag at the foot and the sag at the head for the conditions stated is probably not far from 4 to 1 for the same spacing of troughing idlers.

The spacings given in Table 4-F are not based on theory; they have been found satisfactory in practice for most conveyors where the belt runs straight, does not sag too much, the edges do not flatten down perceptibly, and the loads keep a fairly uniform cross section as they travel along. The table does not distinguish between horizontal and inclined conveyors, long and short ones, but conveyors vary greatly in these respects, and the belt tension in any particular conveyor may vary much as between head tension and foot tension, no load and full load, or even occasional overload. Hence the figures of Table 4-F are to be used with some judgment. Sometimes, instead of having spacing uniform, it will be possible and even advantageous to space the idlers at varying distances, closer together where the belt tension is least and farther apart where the tension is greatest. Page 106 tells why it is proper to put idlers closer together at the foot of a conveyor; that is, the belt will sag less, is less likely to run off to one side and take a lopsided load, and is not so likely to sag or chafe under the skirt boards. As regards the spacing of idlers toward or at the head end, there are factors other than belt sag to be considered: (1) There must be no side spill of material as the belt flattens out in going from the last troughing idler on to the head pulley. (2) The load must not change its cross section between idlers; that is, the edges of the belt must not flatten down (see page 106). (3) The load on each troughing idler must not be too great.

With these limitations in mind let us assume a 36-inch belt conveyor, 400 feet centers, carrying material that weighs 75 pounds per cubic foot under conditions such that the maximum belt tension under full load is 4000 pounds, under no load 2000 pounds, and the tension at the foot end 1000 pounds. Say that from Table 4-F we take 48 inches as the *average* spacing of the idlers corresponding to the *average* pull on the middle of the conveyor, which is 2500 pounds $\left(\frac{4000 + 1000}{2} = 2500\right)$. Then for 1000 pounds tension at the foot the spacing might be as low as $48\sqrt{\frac{1000}{2500}}$ = about 30 inches and at the drive end $48\sqrt{\frac{4000}{2500}}$ = about 60 inches for an equal amount of sag in the belt. For 2000 pounds empty belt pull the span might be $48\sqrt{\frac{2000}{2500}}$ = about 44 inches for the same amount of belt sag. The conveyor described

would require 100 troughing idlers at 48 inches uniform spacing, and if we wish to use the same total number, they could be arranged in four groups thus:

25 at 32 in.	= 800 in.
25 at 48 in.	= 1200 in.
25 at 52 in.	= 1300 in.
25 at 60 in.	= 1500 in.
Total	= 4800 in. = 400 ft.

It might be thought that 60 inches is too great a spacing for some conditions of loading and belt operation, or that the belt might sag too much between idlers when running empty. The second consideration is not so important as the first, but, assuming that a maximum spacing of 54 inches is not too great to meet the three conditions stated above, we may call 54 inches the maximum idler spacing for this conveyor, and 30 inches the minimum, and divide the idlers into, say, five groups with a uniform increment of spacing from foot to head thus:

5 at 30 in.	= 150 in.
5 at 36 in.	= 180 in.
10 at 42 in.	= 420 in.
45 at 48 in.	= 2160 in.
35 at 54 in.	= 1890 in.
100 idlers Total	= 4800 in. = 400 ft.

For spacing of idlers at head end see page 122.
For spacing of idlers at hump see page 124.

These principles have been used in the design of a number of 48-inch conveyors carrying run-of-mine coal at 500 feet per minute. Table 4-G shows the way the troughing idlers are spaced on these conveyors; the final spacing on each conveyor was determined by trial after months of operation. The edges of the belts were carefully watched, and if it appeared that the sag of the belt was enough to cause the load to change its cross section at any place, the idler spacing at that point was reduced to lessen the sag. Such changes brought a considerable reduction in the horsepower readings. On this point see page 121.

One of the considerations stated above as limiting the distance between troughing idlers is an important one. The facts are stated below and illustrated in Fig. 2-19. With 20° troughing the edge of a belt drops less than with 45° troughing; nevertheless, there is still some deflection if the belt carries a full load. When the trough flattens there is a rearrangement of the load on the belt and a resulting squeeze

when the belt is troughed again. The power required for this squeeze may not be much as measured in pounds at each idler, but it causes the friction of the material on itself, and when it is repeated a hundred or more times per minute at each idler, the waste of power is considerable. For instance, in a conveyor 675-foot centers with idlers

TABLE 4-G

GRADUATED SPACING OF IDLERS

Number of spacings of specified lengths

Conveyor Length, feet	Spacing, inches							
	54	51	48	45	42	39	36	Other
1512	82	65	77	71	69	.	8	7 @ 33 in.
1324	72	60	65	64	63	..	2	6 @ 33 in.
1295	65	61	64	62	58	5	3	7 @ 33 in.
1365	69	64	67	67	64		3	9 @ 33 in.
1242	60	60	63	61	57		4	7 @ 33 in.
1846	78	80	82	78	78	76	.	4 @ 30 in.
1675	70	70	74	71	71	73	...	4 @ 30 in.

spaced 3 feet 6 inches apart there are 192 idlers, and at 385-feet-per-minute travel, a given cross section on the belt changes its shape $\frac{385}{3.5} = 110$ times per minute. Since there are 192 places where the trough simultaneously flattens out, the result is that in 1 minute 21,120 squeezes are exerted on the belt and on the material on it, to bring it back to troughed form again.

Some of the power spent in troughed belt conveyors goes for this work. Where the spacing is close, the change of shape is not great, but if the span is too great, the effect is to hurt the belt and waste power.

Tests to Show Effect of Increased Spacing. In a 48-inch belt conveyor with ball-bearing idlers, experiments were made to show the effect of spacing the idlers farther apart. An idler was mounted on a balanced frame and the reaction was weighed by a spring balance. When the spacing was changed from 3 feet to 4 feet the load of coal did not change its shape on the belt, and the reaction at the idler became about one-third greater. When the idler was made to carry 7 or 8 feet of the loaded belt, the load of coal changed its shape perceptibly; it flattened out and some pressure was required to squeeze

it back into troughed form. This pressure exerted by the inclined troughing rolls caused a reaction which showed on the spring balance. It was considerably more than the proportionate increase in the idler spacing.

These experiments have been confirmed by a test of a 48-inch level conveyor 1500 feet centers with ball-bearing idlers spaced about 42 inches apart and carrying 1400 tons of coal per hour. When every other idler was laid down, making the average spacing 7 feet, recording instruments showed an increase of 20 per cent in the power consumed.

These tests show that the saving in cost of troughing idlers which might be effected by increasing the idler spacing is offset by the cost of the power required to squeeze the belt and its load back into troughed form when the spacing becomes too great.

Spacing of Idlers at Loading Point. It is customary to keep the foot wheel about 3 feet back of the first troughing idler so that the belt can assume the fully troughed shape before the load comes on it. The rim of the foot pulley should be level with the horizontal roll of the first idler to avoid the chance that the belt might lift and be damaged by rubbing against the loading chute or the skirt boards. The first roller is set about 6 inches behind the end of the loading chute or wherever the material first strikes the belt, so that the idler is not hurt by the impact of the material, nor is any damage done to the belt where it is backed up by the idler. If the belt is unsupported where the material hits it, it is less likely to be cut or bruised. The spacing of idlers under the skirt boards should be close enough to keep the belt from sagging away from the boards or the rubber guard strips; 24 to 30 inches is usual. If the idler spacing is graduated (see page 117) the idlers just ahead of the loading point are set closer than the average spacing and the belt takes the load of material away from the skirt boards without abrupt change of load cross section. It is good practice to use rubber-covered rolls in idlers at the loading point if the impact is severe (see page 96). Another way is to mount the idler stands at the loading point on rubber filler blocks.

Spacing of Idlers at Head End. The distance from the head pulley to the nearest troughing idler is determined by two considerations. (1) It must not be more than the spacing of the idlers which is considered proper for the load on the belt and the tension on the belt. (On this point see "Graduated Idler Spacing," page 117.) (2) It must not be so short that the edges of the belt are stretched too much as the belt loses its troughed shape and flattens down on the

rim of the head pulley. In order to lessen the amount of this stretch, it is customary to set the rim of the head pulley above the line tangent to the tops of the horizontal rolls of the troughing idlers (Fig. 4-28), so that the length of the stretched edge is approximately the hypotenuse of a triangle with base L and height h, where h is about half the depth of the troughing. For a belt of width w troughed at 20° on three-roll idlers the depth of troughing is about $\frac{w}{9}$ and $h = \frac{w}{18}$. Calling the distance from the head pulley to the nearest idler L and limiting the length of the stretched edge to $1.002L$, which stretch corresponds to less than one-fifth of the tension at which it is proper to work belts (Table 3-D), then $L^2 + \frac{w^2}{324} = [1.002L]^2$, from which $L = 0.88w$. That is, for belts troughed 20° on three-roll idlers and with the head pulley set as shown in Fig. 4-28, the edges of the belt will not be overstretched in the act of flattening down if the distance to the nearest troughing idler is at least equal to the width of the belt. For belts up to 42 inches wide, normal idler spacing is likely to be greater than the width of the belt; for belts wider than 42 inches, it is advisable to make $L = w$. Of course most of the belt stretch at the head pulley comes from the working pull in the belt; we are discussing here that slight increment of stretch which comes when the belt flattens down on the head pulley. The purpose is to keep that increment of stretch low in amount.

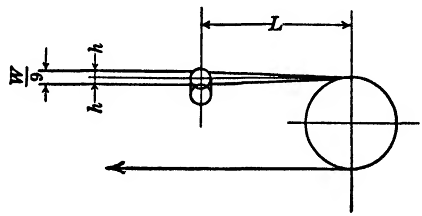


FIG. 4-28. Troughed Belt Flattening at Head Pulley.

Supporting the Belt at Humps or Bends. When a loaded belt runs from an incline to a direction that is less steep or is horizontal, the change of direction can be made over a flat-face pulley or over troughing idlers. If a hump pulley is used, the case is similar to what is discussed above; the rim of the pulley should be set above the tops of the troughing rolls (see Fig. 4-28), and the adjacent idlers should not be too close to the pulley. A safe rule for most cases of this kind is to make the distance L equal to the normal spacing of the idlers on the conveyor, or if the spacing is variable, to the spacing on that part of the length of the conveyor. If a humped conveyor is likely to be stopped and started under heavy load, it might be thought advisable to put troughing idlers close to the hump pulley in order to prevent side spill of material, but it would be better then not to

use a hump pulley at all, but carry the belt over the bend on a group of troughing idlers.

Troughing Idlers at Bends. If a belt is carried over a bend on a group of troughing idlers which are set on an arc of radius R (Fig. 4-29) the edges of the belt travel on an arc of radius larger than R and are stretched. A belt of width w has about $\frac{w}{3}$ in contact with each inclined roll, and since sin 20° is about $\frac{1}{3}$, the depth of troughing on standard idlers is about $\frac{w}{9}$. The radius of the stretched edges is therefore $R + \frac{w}{9}$, and for an angle of bend of a degrees, the stretch of the edges is $2\pi\left(R + \frac{w}{9}\right)\frac{a}{360} - 2\pi R\frac{a}{360}$, which equals $2\pi\frac{w}{9}\frac{a}{360}$. From Table 3-D, on page 57, we take the percentage of stretch due to the

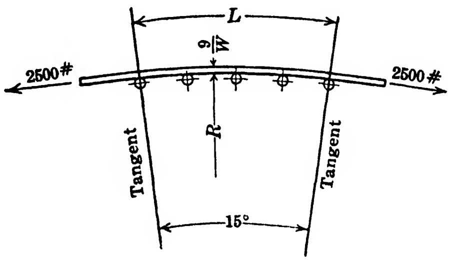


FIG. 4-29. Belt Supported on Idlers at Hump.

unit working tension in the belt, then assume a larger permissible percentage which will include the stretch due to going over the bend. From the difference between the two, we can calculate L, the length of belt on the bend, and from that the radius of curvature R.

Example: If a 36-inch 5-ply belt of 28-ounce duck is under 2500 pounds working tension due to the load at a 15° bend, the unit stress is 14 pounds per inch per ply. The normal stretch due to that stress is 0.75 per cent. If we say that the maximum stretch at the edges of this belt shall not exceed 1.75 per cent, then the stretch in the edges due to the bend alone can be $1.75 - 0.75 = 1$ per cent, which from the table would correspond to 14 pounds per inch per ply. As a matter of fact it will not be so much, because tension at the edges is distributed to some degree throughout the width of the belt, but assuming that the stretch of 1 per cent due to the bend is concentrated at the edges, we say $0.01L = 2\pi\frac{w}{9}\frac{a}{360}$, an equation which for this example gives $0.01L = 2\pi\frac{36}{9}\frac{15}{360}$, from which $L = 104.7$ inches and $R = 401$ inches.

The troughing idlers at a bend should be close enough to support

the belt and its load properly. From Table 4-F on page 118, we see that the spacing need not be less than 42 inches for any load on a 36-inch belt if the idlers are good enough, but another consideration is that the number of idlers on the bend must be enough to take easily the radial reaction due to the pull in the belt. In the present example, the radial reaction for 2500 pounds on a 15° bend is $2500 \times 2 \sin 7\frac{1}{2}°$ or about 650 pounds. If the length of arc 105 inches is divided into four parts of 26 inches each, five idlers will be required, and each will be loaded with, say, 300 pounds of belt and load plus $\frac{650}{5}$ pounds of radial reaction, a total of 430 pounds. If the arc is divided into three parts of 35 inches each, four idlers will be required, the load on each may be assumed to be 450 pounds from the belt and load plus 170 pounds radial reaction, a total of 620 pounds.

The ordinary troughing idler for a 36-inch belt has three rolls of 5- or 6-inch diameter, each with two roller bearings (or ball bearings). A usual rating for a $\frac{3}{4}$-inch roller bearing is 300 pounds radial load at 400 r.p.m. with a life expectancy of 60,000 hours. These figures may be reduced to cover such contingencies as improper setting of idlers, poor lubrication, and accident, and still the margin of safety for a group of four idlers would be plenty for the 36-inch belt under consideration.

Comparative Cost of Hump Pulley or Bend Idlers. Table 4-H shows the approximate cost of a pulley with its accessories as compared with the cost of troughing idlers of standard make. If a 16-inch-diameter pulley for an 18-inch belt costs 4.5 times as much as a troughing idler with 5-inch rolls and 3.75 times as much as one with 6-inch rolls, it appears that a group of four idlers will be less expensive than a hump pulley. For wider belts the ratios are larger and more idlers can be put on a bend at less expense than for a hump pulley.

TABLE 4-H

	Comparative Cost		
	18-in. Belt	30-in. Belt	48-in. Belt
16-in. pulley with shaft, bearings, collars . .	4 5	5 2	6 6
One troughing idler with three 6-in. rolls .	1 2	1 3	1.3
One troughing idler with three 5-in. rolls. .	1 0	1 0	1.0

Belts Supported on Runways. In machines like graders, tunneling machines, and "mucker" machines, where short belts carry the excavated material away from the machine to a dump or to cars, it is often not convenient to support the belt on idlers or rollers because of small space or headroom or difficulty of lubrication or because the impact of large rocks might break or damage the idlers. In such cases belts are made with very heavy rubber covers $3/8$ or $1/2$ inch thick, and travel on the bottom of steel plate troughs. In some machines of this kind used in tunnel building, the rocks are 2 foot cube or larger, the belts run on three shifts a day, and although they do the work, they do not last long. When a new belt must be put on, it is considered merely one of the items of expense of running the machine.

For package-handling belts see page 258.

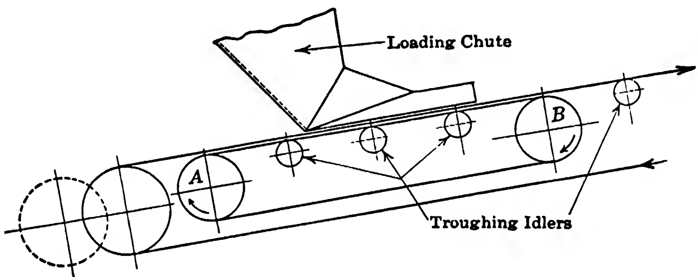


FIG. 4-30. Auxiliary Belt at Loading Point.

Auxiliary Belt at Loading Point. Fig. 4-30 shows a scheme for cushioning a conveyor belt from blows at the loading point. A short belt A–B runs under the conveyor belt and over a few troughing idlers to provide a double thickness of belt where the lumps strike. It was not a success; it cost more than a few rubber-covered rolls, and it was hard to keep the auxiliary belt in its proper place.

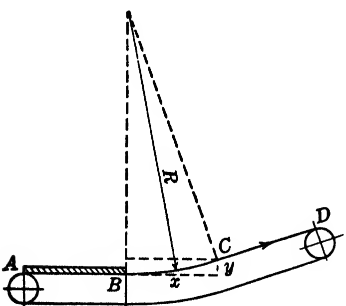


FIG. 4-31. Vertical Curve.

Weight of Conveyor Pulleys. Table 4-J gives the approximate weight of cast-iron pulleys.

Vertical Curves in Belt Conveyors. Fig. 4-31 represents a belt conveyor AD partly inclined and partly horizontal, or at two different angles of inclination. The problem is to find a radius R of the curve BC joining

the two straight portions which is large enough to prevent the belt from lifting off the idlers in section BC under the most unfavorable

TABLE 4-J

APPROXIMATE WEIGHT IN POUNDS OF CAST-IRON CONVEYOR PULLEYS

Diameter in Inches	Face in Inches									
	18	20	22	26	32	38	44	51	57	63
12	110	125	140	160	200	240	290	335		
16	165	190	210	250	310	370	450	520		
18	195	210	245	290	350	400	500	580		
20	220	240	270	310	380	460	570	690	730	770
24	310	330	350	390	520	590	740	865	970	1070
30	400	430	480	560	710	810	1040	1180	1360	1480
36	510	580	630	750	890	1080	1320	1540	1710	1870
42	650	730	790	930	1110	1310	1590	1890	2110	2310
48	....	....	....	....	1470	1710	2110	2440	2730	3030
54	....	....	..	....	1710	2140	2480	2880	3210	3550
60	....	....	....	....	2370	2820	3200	3830	4290	4670

condition, namely, when the section AB is fully loaded and the rest of the conveyor is empty.

The path of the belt, unsupported between B and C, forms part of a catenary, the equation being $y = \frac{x^2}{2a}$ (Marks' "Mechanical Engineers' Handbook," first edition, page 148), where

$$a = \frac{T}{w}$$

T = tension in pounds in the belt at point B (horizontal component of the tension in the catenary).

w = weight of empty belt in pounds per foot of length.

x, y = horizontal and vertical ordinates of the curve.

Then

$$\frac{x^2}{2y} = \frac{T}{w}$$

When $\frac{T}{w}$ is large, as in belt conveyors, the arc BC of the catenary is very flat and for practical purposes may be considered an arc of a circle.

The tangent at point C is $\frac{2y}{x}$, also $\frac{x}{R - y}$. Since the angle at which belt conveyors operate varies between 0° and 25°, the expression $\frac{x}{R - y}$ will vary between $\frac{x}{R}$ for 0° to $\frac{x}{0.9R}$ for 25° .

Assuming the steep angle, then

$$\frac{2y}{x} = \frac{x}{0.9R} \qquad \text{or} \qquad 0.9R = \frac{x^2}{2y}$$

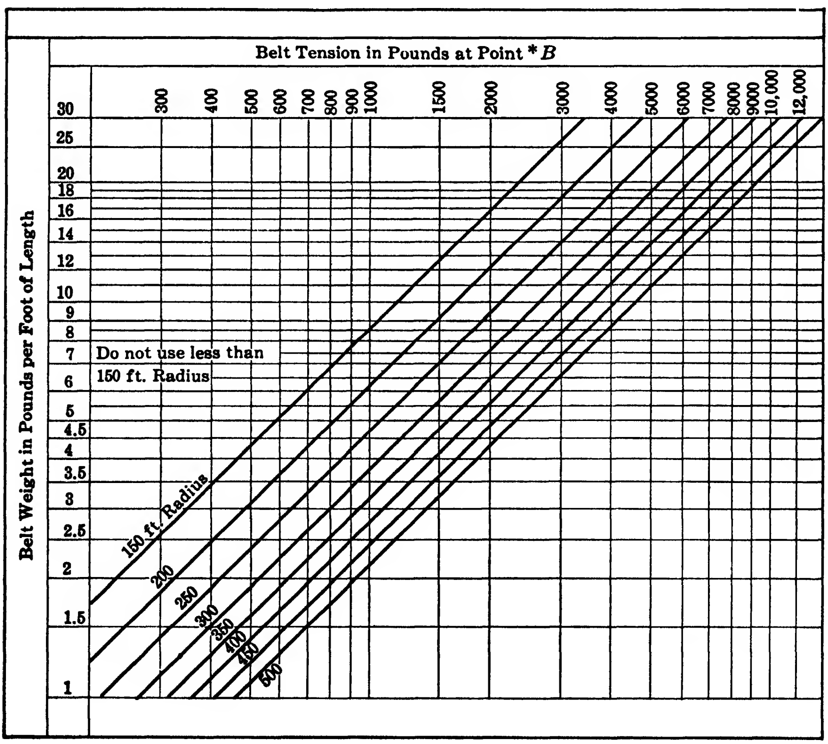
Therefore

$$0.9R = \frac{T}{W} \qquad \text{and} \qquad R = 1.1\,\frac{T}{W}$$

Considering all the contingencies of belt conveyor practice it is safer

TABLE 4-K.

RADII FOR VERTICAL CURVES.



to increase R rather than decrease it. Table 4–K gives valucs of R when T and w are known based on:

$$150\text{-ft. radius} = 1.25\,\frac{T}{w}$$

$$300\text{-ft. radius} = 1.15\,\frac{T}{w}$$

$$500\text{-ft. radius} = 1.1\,\frac{T}{w}$$

CHAPTER 5

DRIVING THE BELT

Horsepower to Drive Belt Conveyors. In general,

$$HP = \frac{\text{Effective pull in pounds} \times \text{Speed in feet per minute}}{\text{33,000 foot-pounds per minute}}$$

In a belt conveyor the output of the motor is expended in overcoming transmission losses up to the conveyor drive pulley and in putting a certain pull into the belt. The transmission losses will be considered later; the effective belt pull, which is a measure of the horsepower used between the terminal pulleys of the conveyor, may be considered as made up of several items.

1. The power required to drive the conveyor empty, the amount being practically the same whether the conveyor is horizontal or inclined.

2. The power required to move the material horizontally.

3. The power required to move the material vertically, a value which may be positive or negative.

Power required to run the belt empty over and above what is due to air resistance is composed of the power required to:

1*a*. Revolve the rolls of the idlers with the belt weight on them.

1*b*. Bend the empty belt into troughed form (unless it runs flat).

1*c*. Revolve the terminal and intermediate shafts.

1*d*. Bend the belt around the pulleys and as it flexes up and down in passing over the idlers.

Power required to move the material horizontally can also be subdivided; it is due to:

2*a*. Increased effort to revolve the idler rolls due to the weight of the material.

2*b*. Additional force required to maintain the mass of material in a troughed form (unless the belt runs flat).

2*c*. Added force to revolve the terminal shafts in their bearings due to the greater tension in the belt.

2*d*. Added force to bend the belt over the pulleys and idlers due to greater tension in the belt.

2e. Power to accelerate the material at the loading point, that is, to bring it from chute speed up to belt speed.

Power required to move the material vertically is made up of:

3a. Power to move vertically a weight at a given speed; in an inclined conveyor a plus value for lifting or a minus value for lowering the load.

3b. Added power to revolve the terminal shafts due to additional belt tension, a plus value in an inclined conveyor.

3c. Added power to bend the belt around the pulleys due to greater tension in the belt, a plus value in an inclined conveyor.

Of all the separate items listed above, it is impossible to assign theoretical values to any except 3*a*, which concerns lifting and lowering the load. It is difficult to determine in any conveyor, by direct means, what the separate friction losses are. Items 1*b* and 1*d*, troughing the empty belt and bending the belt around the pulleys, are greatest when the belt is first installed and gradually decrease over a period of time as the belt "limbers up." However, from tests showing power readings of many conveyors it is possible to make charts which give the information indirectly, that is: (1) by showing the form which a horsepower formula should take; (2) by indicating closely enough for practical purposes the values of the variables and constants in such a formula.

Tests of Belt Conveyors. Tests were made of eighteen 48-inch belt conveyors, all alike in design, carrying 1700 tons of coal per hour at 490 feet per minute. Idler rolls were of 7-inch diameter and had live shafts mounted on ball or roller bearings. Belt and moving parts of idlers weighed 67 pounds per foot of conveyor. All machinery shafts were carried on roller bearings; all equipment was first class, and maintenance was excellent.

Table 5-A records in column 3 the power required to run these conveyors empty. The recorded horsepowers for the conveyors loaded were adjusted to eliminate the power for elevating coal in those conveyors which ran uphill and for lowering it in those which ran downhill. The conveyors were in tandem and were inclined up or down to follow the rise and fall of the mine passages in which they were installed. These adjusted values, given in column 5, represent what the horsepowers would have been had the conveyors been level and had the transfer of coal been horizontal. The differences between column 3 and column 5 show the amount of power expended in the horizontal transfer only. They are given in column 4.

These results are shown in diagram form in Fig. 5-1. Conveyor lengths are laid off on the horizontal, horsepowers as verticals; line *A* shows how the horsepowers for the empty conveyors varied with the

length, and line *B* shows how the horsepowers for horizontal transfer of material varied with the lengths. The ordinates of line *A* measured

TABLE 5-A

TEST HORSEPOWERS OF 48-INCH CONVEYORS

1	2	3	4	5	6
Conveyor	Centers in feet	HP. Empty	HP. Material Moved Horizontally	HP. Total	Weight of Terminal Pulleys in pounds
C-3	319	10 5	15 7	26 2	21,312
C-2	415	11 4	18.6	30 0	21,318
C-1	779	17 0	29 6	46 6	30,684
C-4	1027	19 2	37 1	56 3	32,312
C-5	1100	19 9	39 3	59 2	30,834
C-19	1242	22 3	44.6	66 9	31,744
C-16	1262	22 2	45 2	67 4	29,330
C-15	1295	22 7	46.2	68 9	30,834
C-18	1300	22 8	46 4	69 2	30,834
C-12	1318	23 0	46 9	69 9	30,831
C-13	1324	23 1	47 1	70 2	30,834
C-14	1341	23 1	47 6	70 7	29,330
C-17	1365	23 6	48 4	72 0	30,834
C-7	1401	23 8	49 5	73 3	29,330
C-10	1408	24 1	49 7	73.8	30,834
C-6	1495	25 1	52 4	77 5	30,834
C-11	1512	25 4	52 9	78 3	30,834
C-8 & 9	2435	36 6	81 5	118 1	29,327

DATA

Forty-eight-inch-wide belts running at 490 ft per min
Weight of belt and moving parts of idlers, 67 lb per ft. of conveyor.
Capacity, 1700 tons per hour of coal, 2000 lb per ton.
Diameter of idler rolls, 7 in
Roll shafts turn in ball bearings mounted in pivotally supported housings.
Terminal shafts turn in anti-friction bearings
Highest-grade equipment and excellent maintenance

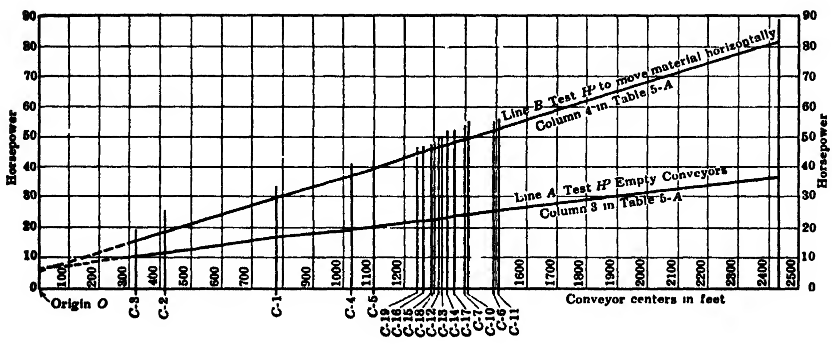


FIG. 5-1. Diagram of Test Horsepowers of 48-inch Conveyors.

above the line 0-0 indicate the power required to overcome the sum of the resistances 1*a*, 1*b*, 1*c*, 1*d*; the ordinates measured above the line 0-0 of line *B* show the power for the resistances 2*a*, 2*b*, 2*c*, 2*d*.

Tests were also made of four 60-inch conveyors all alike as to design, carrying 1960 tons of coal per hour at 545 feet per minute. Idler rolls were 8 inches in diameter and had live shafts mounted in

TABLE 5-B

TEST HORSEPOWERS OF 60-INCH CONVEYORS

1	2	3	4	5	6
Conveyor	Centers in feet	HP. Empty	HP. Material Moved Horizontally	HP. Total	Weight of Terminal Pulleys in pounds
P-1	350	18.0	19.6	37.6	35,760
P-3	475	19.9	24.1	44.0	35,940
P-4	1844	44.0	72.8	116.8	35,800
P-5	2217	51.2	86.1	137.3	37,800

DATA

Sixty-inch-wide belts running at 545 ft per min
Weight of belt and moving parts of idlers, 99.2 lb per ft of conveyor.
Capacity, 1960 tons per hour of coal, 2000 lb per ton
Diameter of idler rolls, 8 in.
Roll shafts turn in ball bearings mounted in pivotally supported housings
Terminal shafts turn in anti-friction bearings.
Highest-grade equipment and excellent maintenance.

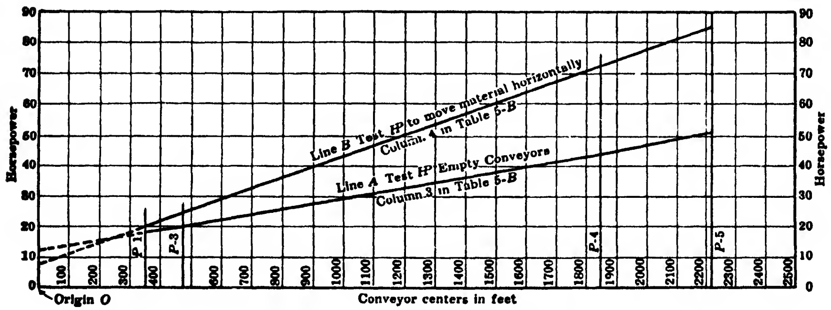


FIG. 5-2. Diagram of Test Horsepowers of 60-inch Conveyors.

ball bearings. Belt and moving parts of idlers weighed 99 pounds per foot of conveyor. All machinery shafts were carried in roller bearings; all equipment was first class, and maintenance was excellent.

Table 5-B and Fig. 5-2 record the facts for these four 60-inch conveyors just as Table 5-A and Fig. 5-1 do for the eighteen 48-inch conveyors.

In 1939 tests were made of a tandem series of 24-inch belt conveyors carrying crushed rock on commercial anti-friction idlers with 6-inch rolls. Horsepower diagrams based on them (not shown here) were similar to those shown in Figs. 5-1 and 5-2.

From these tables and figures we derive the following conclusions:

1. The relation between conveyor length and horsepower shown in the diagrams is a straight line.
2. The line does not pass through the origin point O.
3. Each line can be represented by the general equation of a straight line: $y = mx + C$.
4. The slope or angle that the horsepower line makes with the horizontal depends on some friction factor and the rate at which the friction is overcome, that is, the rate at which work is done.

These tests and records of other tests form the basis of the values shown in Tables 5-E and 5-F.

Development of a Formula for Horsepower. A graphical representation of the straight-line formula

$$y = mx + C$$

is shown in Fig. 5-3; here y = horsepower, x = conveyor length in feet, m = tangent or rate of change in horsepower as the conveyor length changes, C = horsepower required that is independent of conveyor length.

Horsepower for the Empty Conveyor. If Q = weight in pounds of the belt and moving parts of idlers per foot of conveyor and F = friction factor, then the pull required to move 1 foot of conveyor is FQ and the horsepower per foot of conveyor is $\frac{FQS}{33{,}000}$. Here QS is in pounds per minute, but for convenience in combining it with formula 2 for the material carried, we will express it in tons per hour. Then

$$\frac{QS \times 60 \text{ minutes}}{2000} = \frac{QS}{33} = B$$

where B is an expression for the weight of the belt and the moving parts of idlers per foot of conveyor combined with the belt speed of S in feet per minute as tons per hour. Substituting $B = \frac{QS}{33}$ in $\frac{FQS}{33{,}000}$, we have for the horsepower for 1 foot of empty conveyor

$$HP = 0.001\,FB$$

since 0.001 FB is the horsepower for each foot of empty conveyor;

it is the rate of change in horsepower and corresponds to m in the straight-line formula.

The value of F, the friction factor, is derived from horsepower tests. In Fig. 5-1 or in Fig. 5-2 the angle of line A can be measured or calculated and the tangent of the angle so determined equals 0.001 FB and $F = \frac{\text{tangent}}{0.001\ B}$. If the line A passed through the origin point O, the expression for the horsepower for the empty conveyor would be simply $HP = 0.001\ FBL$, where L is the length of the conveyor in feet; but the line A passes at a distance above the origin, and for the amount of horsepower indicated by this distance it is necessary to add something which corresponds to C in the straight-line formula. This added horsepower is independent of the length of the conveyor; it is related to the weight of the machinery of the conveyor, and ultimately to the weight of the belt and the moving parts of the idlers stated as pounds per foot of conveyor. In Fig. 5-1 or Fig. 5-2 the vertical distance of line A above the origin is measured in terms of horsepower and called eB, or $e = \frac{\text{Added } HP}{B}$. The formula for the horsepower for the empty conveyor then becomes

$$HP = 0.001\,FBL + eB \tag{1}$$

It is shown graphically in Fig. 5-4.

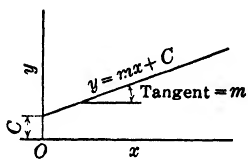


Fig. 5-3.

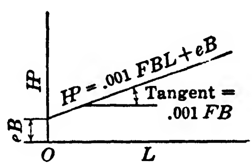


Fig. 5-4.

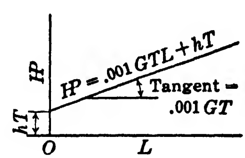


Fig. 5-5.

Graphical Representation of Equations:

Fig. 5-3. General Equation of a Straight Line.
Fig. 5-4. Horsepower to Operate Empty Conveyor.
Fig. 5-5. Horsepower to Move the Material Horizontally.

Horsepower Required to Carry Material Horizontally over and above That for the Empty Conveyor. If T = tons per hour of material moved (2000 pounds per ton), G = friction factor, and S = belt speed in feet per minute, then the weight of material in pounds on each foot of conveyor is $\frac{T \times 2000 \text{ lb.}}{60 \text{ minutes} \times S}$ or $\frac{33.3\ T}{S}$ and the horsepower

for each foot of conveyor is $\frac{GT\,33.3}{S} \times \frac{S}{33{,}000}$ or 0.001 GT. Since this is the horsepower for each foot of conveyor it is the rate of change of horsepower and it corresponds to m in the straight-line formula.

The value of G, the friction factor for the load on the belt, is derived from horsepower tests. In Fig. 5–1 or in Fig. 5–2 the angle of line B can be measured or calculated and the tangent of the angle so determined equals 0.001 GT, and $G = \frac{\text{Tangent}}{0.001\ T}$. If the horsepower line B passed through the origin point O, the expression for the horsepower to move material horizontally would be simply $HP = 0.001\ GTL$, where L is the length of the conveyor in feet; but the line B passes at a distance above the origin, and for the amount of horsepower represented by this distance it is necessary to add something which corresponds to C in the straight-line formula. This added horsepower is independent of the length of the conveyor; it is related to the tonnage carried by the conveyor and is represented by power required for the terminals of the conveyor.

In Fig. 5–1 or 5–2 the vertical distance between line B and the origin is measured in terms of horsepower and called hT or $h = \frac{\text{Added } HP}{T}$. The horsepower required to move material horizontally over and above the power to operate the conveyor empty then becomes

$$HP = 0.001\ GTL + hT \tag{2}$$

It is shown graphically in Fig. 5–5.

Summary. The following shows how formulas 1 and 2 correspond with the equation of a straight line.

Straight line	y	=	m	x	+	c
Formula 1	HP	=	$0.001\ FB$	L	+	eB
Formula 2	HP	=	$0.001\ GT$	L	+	hT

Combining 1 and 2, we have

$$\text{Total } HP = 0.001\ L(FB + GT) + eB + hT \tag{3}$$

which is the general formula for the horsepower for a loaded horizontal conveyor.

Use of Formula 3. To find the horsepower for a belt conveyor of length L in feet and a capacity T in tons per hour, it is necessary to know B, F, e, G, and h for the particular conveyor. Since $B = \frac{QS}{33}$ (see page 134), it is possible to show in a table values of B at an

assumed belt speed if the weight of belt and moving parts of idlers is known. In Table 5–C, belt speed is assumed to be 100 feet per minute, idlers are commercial anti-friction style with 6-inch rolls, idler spacing is that which is ordinarily used, and the belt is of an average weight likely to be used with those idlers.

TABLE 5–C

VALUES OF B

Belt Width, inches	Q, pounds	B
14	15	45
16	16	48
18	19	57
20	21	63
24	25	75
30	34	102
36	44	132
42	50	150
48	60	180
54	69	207
60	77	231

The values of B shown in the table are accurate enough for most conveyors and for preliminary calculations. But for conveyors where the belt or idler construction or idler spacing differs from normal practice or when it is desired to improve the accuracy of the results, it is advisable to determine Q and B more accurately for that particular conveyor.

Example. Assume a 36-inch 6-ply belt of 32-ounce duck with ⅛-inch and 1/16-inch covers, troughing idlers with 6-inch rolls, spaced 54 inches, return rolls spaced 10 feet apart, speed 375 feet per minute.

Belt weight per foot	9 lb.	Table 3–E, page 59
Troughing idler.	58 lb.	Table 4–E, page 116
Return idler.	46 lb.	Table 4–E, page 116

Hence Q equals,

for belt	9×2	$= 18$ lb.
for troughing idler	$58 \div 4.5$	$= 12.9$ lb.
for return idler	$46 \div 10$	$= 4.6$ lb.

a total of 35.5 pounds per foot of conveyor; and, since the speed is not 100 feet per minute, but 375 feet,

$$B = \frac{35.5 \times 375}{33 \times 100} = 403$$

The values of the factors F, e, G, and h in formula 3 must be derived from tests of conveyors in use. Those given in Table 5–D are based on the conditions stated in the first column.

TABLE 5-D

FACTORS FOR FORMULA 3

	F	e	G	h
1. Plain bearing idlers, all machinery shaft bearings babbitted, average operating conditions .	0 056	0.004	0.073	0.002
2. Commercial anti-friction idlers with 5-in. or 6-in. rolls, all machinery shaft bearings babbitted, average operating conditions	0 028	0 004	0.042	0.002
3. High-grade anti-friction idlers with 7-in. or 8-in. rolls, all machinery shafts with anti-friction bearings, good operating conditions. Load cross section on belts larger than ordinary. Loading conditions excellent. See Fig. 5–1...	0 015	0 007	0 022	0.004
4. Highest-grade anti-friction idlers with 8-in. rolls, all machinery shafts with anti-friction bearings, operating and loading conditions the best possible. Load cross section on belts the largest possible. Tests made after the belts and machinery had been "run in" for a year or more. See Fig. 5–2 .	0 012	0 007	0 018	0 004

The friction factors F and G in Table 5–D are not merely coefficients of rolling friction. F covers not only item 1*a* (page 130) but also 1*b* and 1*d*. For the loaded conveyor, G covers not only 2*a* (page 130) but also 2*b* and 2*d*. Since 2*b* is larger than 1*b*, G is larger than F. The numerical value of G depends on the weight and nature of the material, the cross section of the load on the belt, the spacing of the idlers, and the thickness and stiffness of the belt. It is not practicable to assign to G variable values for all materials, loadings on the belt, spacing of idlers, and belt stiffness. The values given in Table 5-D, lines 1 and 2, represent average practice of the present day, that is, loads with 20° surcharge; Table 8-B, belt thickness and weight of duck according to Table 6-A, and idler spacing according to Table 4-F. If in any conveyor these items differ much from the assumptions stated, it is proper to use higher or lower values for B, Table 5-C,

and for G in Table 5-D. An example of this is shown in Tables 5-G and 5-H giving the horsepowers of grain conveyors.

Lines 3 and 4 in Table 5-D give values which have been derived from tests of large conveyors under very favorable conditions; Tables 5-A and 5-B. They do not apply to ordinary commercial conveyors.

Advantage of Two Friction Factors. In the past it has been common practice to use one friction factor for both the dead and live loads. If the ratio between the dead and live loads were always the same, one average factor could be selected that would give the correct result for a loaded horizontal conveyor.

However, if it is desired to separate the horsepower of a loaded conveyor into its two parts, the power for the empty conveyor and the power to move the material, then the use of one average friction factor will give results too high for the empty conveyor and too low for moving the material.

The use of two friction factors improves the accuracy of the results when calculating the power required for an empty conveyor or the power required to move only the material.

If the size of lumps handled determines the width of the belt,

TABLE 5-E

NET HORSEPOWER AT DRIVE SHAFT—EMPTY CONVEYOR

Belt runs over commercial anti-friction idlers, 3 rolls, 5 or 6 inches in diameter.
Formula is $HP = 0.001 \times 0.028BL + 0.004B$.
Belt speed 100 feet per minute.
For other speeds multiply by $\dfrac{\text{Speed } FPM}{100}$.

Belt Width, inches	Length of Conveyor, feet, center to center											
	25	50	100	150	200	250	300	400	500	750	1000	1500
14	0.21	0.24	0.31	0.37	0.43	0.50	0.56	0.68	0.81	1.13		
16	0.23	0.26	0.33	0.39	0.46	0.53	0.60	0.73	0.86	1.20		
18	0.27	0.31	0.39	0.47	0.55	0.63	0.71	0.87	1.03	1.43	1.82	
20	0.30	0.34	0.43	0.52	0.60	0.69	0.78	0.96	1.13	1.58	2.02	
24	0.35	0.41	0.51	0.62	0.72	0.83	0.93	1.14	1.35	1.88	2.40	3.45
30	0.48	0.55	0.69	0.84	0.97	1.12	1.26	1.55	1.84	2.55	3.26	4.69
36	0.62	0.71	0.90	1.08	1.27	1.45	1.64	2.01	2.38	3.30	4.22	6.07
42	0.71	0.81	1.02	1.23	1.44	1.65	1.86	2.28	2.70	3.75	4.80	6.90
48	0.85	0.97	1.22	1.48	1.73	1.98	2.23	2.74	3.24	4.50	5.76	8.28
54	0.97	1.12	1.41	1.70	1.99	2.28	2.57	3.15	3.73	5.18	6.62	9.52
60	1.09	1.25	1.57	1.89	2.22	2.54	2.86	3.51	4.16	5.78	7.39	10.63

and the tonnage is not great enough to load the belt fully at the speed at which it is desirable to operate it, the ability to separate the power for the empty conveyor from the power to move the material is an advantage.

Table 5-E gives in form for quick reference horsepowers for empty belts at an assumed speed of 100 feet per minute over commercial anti-friction idlers. For grain conveyors refer to Tables 5-G and 5-H.

TABLE 5-F

NET HORSEPOWER AT DRIVE SHAFT—MATERIAL ONLY—MOVED HORIZONTALLY

Belt runs over commercial anti-friction idlers, 3 rolls, 5 or 6 inches in diameter. Formula is $HP = 0.001 \times 0.042TL + 0.002T$.

Tons per Hour	Length of Conveyor, feet, center to center											
	25	50	100	150	200	250	300	400	500	750	1000	1500
50	0 15	0 21	0.31	0.42	0.52	0.63	0.73	0.94	1.15	1.68	2.20	3.25
75	0 23	0 31	0.47	0 62	0.78	0.94	1.10	1.41	1.73	2.51	3.30	4.88
100	0 31	0.41	0.62	0.83	1.04	1.25	1.46	1.88	2.30	3.35	4.40	6.50
150	0 46	0.62	0.93	1.25	1.56	1.88	2.19	2.82	3.45	5.03	6.60	9.75
200	0 61	0.82	1.24	1.66	2.08	2.50	2.92	3.76	4.60	6.70	8.80	13.0
250	0 76	1 03	1.55	2.08	2.60	3.13	3.65	4.70	5.75	8.38	11.0	16.25
300	0 92	1 23	1.86	2.49	3.12	3.75	4.38	5.64	6.90	10.1	13.2	19.5
400	1.22	1.64	2.48	3.32	4.16	5.00	5.84	7.52	9.20	13.4	17.6	26.0
500	1 53	2 05	3.10	4.15	5.20	6.25	7.30	9.40	11.5	16.8	22.0	32.5
750	2.29	3 08	4 65	6.23	7.80	9.38	11.0	14.1	17.3	25.1	33.0	48.8
1000	3 05	4.10	6 20	8.30	10.4	12.5	14.6	18.8	23.0	33.5	44.0	65.0

TABLE 5-G

HORSEPOWER FOR EMPTY GRAIN CONVEYORS

Formula is $HP = 0.001 \times 0.028BL + 0.004B$.

Values given below assume anti-friction idlers and belt speed = 100 feet per minute; for other speeds multiply values by $S/100$.

Belt Width, Inches	Conveyor Length, feet, center to center						*B*
	100	200	300	500	750	1000	
18	0 29	0.41	0.52	0 76	1.05	1.34	42
20	0 31	0 43	0.56	0.81	1.13	1.44	45
24	0 37	0 52	0 67	0 97	1.35	1.73	54
30	0 43	0.62	0 78	1.13	1.58	2.01	63
36	0 51	0 73	0 93	1 35	1 89	2.40	75
42	0 59	0 84	1 08	1 56	2.18	2.78	87
48	0.69	0.97	1 23	1.79	2.48	3.16	99

Table 5-F gives horsepowers for material only, carried horizontally. If the conveyor is inclined, consider what is to be added or subtracted for lifting or lowering the load; see Table 5-J.

TABLE 5-H

HORSEPOWER FOR GRAIN ONLY, MOVED HORIZONTALLY

Formula is $HP = 0.001 \times 0.035TL + 0.002T$.

Values given below assume anti-friction idlers and 1000 bushels per hour of grain weighing 60 pounds per bushel. For other amounts x bushels multiply values below by $x/1000$; for grain weighing w pounds per bushel, multiply by $w/60$.

Bushels per Hour at 60 lb. per Bushel	Conveyor Length, feet, center to center						T
	100	200	300	500	750	1000	
1000	1 65	2 70	3 75	5 85	8 48	11 1	300

Horsepower of Inclined Conveyors. Theoretically, the power to move the conveyor empty should be based on a length equal to the center to center of the terminal pulleys measured on the incline. When the incline is small, say less than 10°, the difference between the inclined length and the horizontal length is small and frequently it is accurate enough to use the horizontal length.

The power required to move the material is divided into two parts: that required to move the material horizontally, and that to move it vertically. To move the material horizontally the power should theoretically be based upon a length equal to the horizontal distance between the terminal pulleys. Since the difference between the horizontal and the inclined lengths is small, particularly when the incline is less than 10°, only a small error results if the inclined length is used. The power to move the material vertically should be based upon the vertical distance from the loading point to the discharge point, which is the top of the end pulley.

Horsepower to Move Material Vertically. If H is the vertical difference in elevation in feet between the loading and discharge points and T is the tons (2000 pounds each) handled per hour, then

$$HP = \frac{T \times 2000 \times H}{60 \times 33{,}000} = \frac{TH}{990}$$

This expression has a plus value when the discharge is higher than the loading point and a minus value when the discharge point is lower

than the loading point, (see 3a, page 131). If the fall or descent of the material is great enough, $\frac{TH}{990}$ may be greater than the horsepower given by formula 3 (page 136); then the net horsepower is negative, and if the conveyor has an electric motor, current will be generated by it.

The power required for items 3b and 3c, page 131, is generally small; for ordinary conveyors with babbitted bearings in the terminal shafts it may be taken as 1 per cent of $\frac{TH}{990}$.

The total horsepower for a loaded inclined conveyor is the value obtained from formula 3, or Tables 5-E and 5-F plus or minus $\frac{TH}{990}$ plus 1 per cent of $\frac{TH}{990}$. For values of $\frac{TH}{990}$ see Table 5-J.

TABLE 5-J

HORSEPOWER TO LIFT MATERIAL VERTICALLY $\left(HP = \frac{TH}{990}\right)$

For horsepower for level conveyors see Tables 5-E and 5-F

Material, tons per hour	Height of Vertical Lift, feet											
	5	10	15	20	30	40	50	60	70	80	90	100
5	0.03	0.06	0.09	0.11	0.16	0.21	0.26	0.31	0.36	0.41	0.46	0.51
10	0.06	0.11	0.16	0.21	0.31	0.41	0.51	0.61	0.71	0.81	0.91	1.01
15	0.09	0.16	0.24	0.31	0.46	0.61	0.76	0.92	1.07	1.22	1.37	1.52
20	0.11	0.21	0.31	0.41	0.61	0.81	1.01	1.22	1.42	1.62	1.82	2.02
25	0.14	0.26	0.39	0.51	0.76	1.02	1.27	1.52	1.78	2.03	2.28	2.53
50	0.26	0.51	0.76	1.01	1.52	2.02	2.53	3.03	3.54	4.04	4.55	5.05
100	0.51	1.01	1.52	2.02	3.03	4.04	5.05	6.06	7.07	8.08	9.09	10.10
200	1.01	2.02	3.03	4.04	6.06	8.03	10.10	12.12	14.14	16.16	18.18	20.20
300	1.52	3.03	4.55	6.06	9.09	12.12	15.15	18.18	21.21	24.24	27.27	30.30
400	2.02	4.04	6.06	8.08	12.12	16.16	20.20	24.24	28.28	32.32	36.36	40.40
500	2.53	5.05	7.58	10.10	15.15	20.20	25.25	30.30	35.35	40.40	45.45	50.50
1000	5.05	10.10	15.15	20.20	30.30	40.40	50.50	60.60	70.70	80.80	90.90	101.0

Horsepower Required for Trippers. In all belt trippers, power is consumed in lifting the material from the level of the belt to the top of the discharge pulley. This power is determined by the tons per hour handled and the lift in feet and can be obtained from Table 5-J. This can be taken as the power required for a fixed tripper. If the tripper is movable, additional power is required to propel the machine along its track. The power for traversing is determined by the weight of the tripper, the coefficient of rolling friction, and the speed at which the tripper is moved. The tripper may be propelled by a small

motor mounted on the tripper frame or by means of a separate winding machine, but if it is moved by the conveyor belt there is increased tension in the belt and some added load on the conveyor drive.

The horsepower to be added, for standard trippers, similar to Fig. 9–15, is:

To lift material:

$$\text{Horsepower} = ZT$$

where Z is the lift in feet for the particular size of tripper divided by 990, and T equals tons per hour passing over the tripper.

To move tripper:

$$\left\{\begin{array}{l}\text{At a speed equal to} \\ \frac{1}{6} \text{ of the belt speed}\end{array}\right\} \quad \text{Horsepower} = YS$$

where Y is the pull in pounds required to move the tripper divided by $33{,}000 \times 6$; and S equals the speed of the belt in feet per minute.

VALUES FOR Z AND Y

	Belt Width in Inches									
	14	16	18	20	24	30	36	42	48	54
Z	0 0035	0 0035	0 0035	0 004	0 004	0 005	0 005	0 0055	0 006	0 007
Y	0 002	0 002	0 0026	0 0029	0 0034	0 0047	0 006	0 0069	0 0082	0.010

Example. What power is required for a standard tripper on a 36-inch wide belt, running 350 feet per minute and carrying 400 tons per hour.

Horsepower to lift material:

$$HP = ZT = 0.005 \times 400 = 2.0$$

Horsepower to move tripper

$$HP = YS = 0.006 \times 350 = 2.1$$

Total horsepower equals 4.1

Power Losses between the Drive Shaft and the Motor. The horsepower calculated from the formulas or taken from Tables 5–E, 5–F, and 5–J is the net horsepower which the head pulley delivers to the conveyor belt. The horsepower output of the motor should be enough larger to allow for losses in the power transmissions between the motor and the head shaft. It is customary to add to the net horsepower 10 per cent for each pair of cast-tooth gears or cast-tooth sprocket wheels, 5 per cent for each pair of cut-tooth gears or cut-

tooth sprocket wheels, 5 per cent for each belt drive, and 3 to 7 per cent for each enclosed speed reducer with spur gears and anti-friction bearings. For a speed reducer fitted with worm gears, the maker should be consulted about what power loss may be expected.

Relation of Horsepower to Belt Tension. Knowing the horsepower to drive a belt conveyor, we can find out the tension in the belt. The horsepower pull, or effective pull, is $\frac{HP \times 33{,}000}{\text{Belt speed in feet per minute}}$ in pounds. This is not the actual or total pull in the belt, because in order to maintain the belt in driving contact with the driving pulley the belt must be kept under tension on both the entering and the leaving sides of the pulley. If in Fig. 5–6 we call the pull on the entering belt T_1 and the pull in the leaving belt T_2, the effective pull in the belt which does useful work or which transmits horsepower is $T_1 - T_2$.

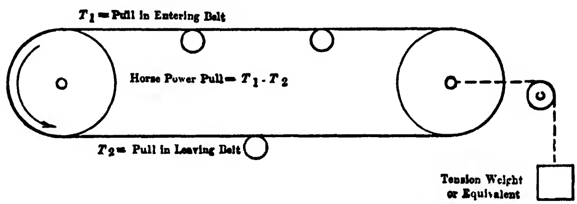


FIG. 5–6. Relation between Horsepower Pull and Belt Tensions.

This gives a difference of belt tensions but not the tensions themselves. From considerations of belt wrap and belt friction we can find the ratio of the tensions $\frac{T_1}{T_2}$, and by combining this with $T_1 - T_2$ we determine the actual values of T_1 and T_2.

Thickness of Belt as Determined by Belt Tension. The total tension T_1 on the pulling side of a belt is made up of the horsepower pull plus an added tension necessary to maintain a driving contact between belt and pulley. The tension T_2 on the leaving side should be only what is necessary to maintain the driving contact. The ratio $\frac{T_1}{T_2}$ depends upon the coefficient of friction between belt and pulley and on the angle in degrees of belt wrap on the pulley. Calling the former f and the latter a, the mathematical expression * is

$$\frac{T_1}{T_2} = 10^{0.00758fa}$$

* For the theory of power transmission by belting, see Kent's *Mechanical Engineers' Handbook* in the Wiley's Engineering Handbook Series, Vol. III, page 24–15.

Experiments have shown that for dry, clean rubber belts on cast-iron pulleys $f = 0.25$, and when the pulleys are lagged or covered with rubber $f = 0.35$. Table 5-K gives values of $\frac{T_1}{T_2}$ for these two coefficients and for various angles of belt wrap.

TABLE 5–K

RATIO OF $\frac{T_1}{T_2}$ FOR VARIOUS CONDITIONS OF DRIVING

T_1 = pulling tension at drive pulley; T_2 = slack tension at drive pulley

Angle of Belt Wrap, degrees	Bare Iron Pulleys	Lagged or Covered Pulleys	Angle of Belt Wrap, degrees	Bare Iron Pulleys	Lagged or Covered Pulleys
135	1 80	2.28	215	2.56	3.72
140	1.84	2.35	220	2 61	3.83
145	1.88	2.43	225	2.67	3 95
150	1 92	2.50	240	2.85	4.33
155	1.97	2 58	255	3.04	4.75
160	2.01	2 66	270	3.25	5.20
165	2 06	2.74	285	3.47	5.70
170	2.10	2.83	300	3 70	6.25
175	2.15	2 91	315	3.95	6 85
180	2.19	3 00	330	4 22	7.51
185	2 24	3 10	345	4 51	8.23
190	2.29	3.19	360	4 80	9.02
195	2.34	3 29	420	6 25	13.00
200	2.39	3 39	500	8.86	21.21
205	2.45	3 50	600	13 71	39.06
210	2 50	3 61	700	21.21	71.96

Since the torsion at the drive shaft is measured by the horsepower pull, which is $T_1 - T_2$, and since the total pull in the belt is T_1, the ratio $\frac{T_1}{T_1 - T_2}$ expresses the relation between the actual effective horsepower pull and the total pull in the belt. These ratios are given in Table 5–L.

From Table 5–L we see that for a simple drive with 180° wrap on an iron pulley $\frac{T_1}{T_2} = 2.193$ or $T_2 = 0.456T_1$; then since horsepower pull = $T_1 - T_2$ horsepower pull = $0.544T_1$ or T_1 = 1.838 horsepower pull (see Table 5–L). That is, the total tension in the pulling side is over 1.8 times what is necessary to move the load and overcome idler friction.

If the pulley is lagged, the ratio falls to 1.5, and if by use of a snub pulley the wrap on the lagged pulley is increased to 240°, the total ten-

sion T_1 becomes only 1.3 times the horsepower pull or effective tension. (See Table 5-L.)

TABLE 5-L

RATIO OF $\frac{T_1}{T_1 - T_2}$ FOR VARIOUS CONDITIONS OF DRIVING

T_1 = tension in pulling belt at drive pulley; T_2 = tension in slack belt at drive pulley; $T_1 - T_2$ = horsepower pull

Angle of Belt Wrap, degrees	Bare Iron Pulleys	Lagged or Covered Pulleys	Angle of Belt Wrap, degrees	Bare Iron Pulleys	Lagged or Covered Pulleys
135	2 25	1 78	215	1 64	1 37
140	2.19	1 74	220	1 62	1 35
145	2 14	1 70	225	1 60	1 34
150	2 08	1 67	240	1 54	1 30
155	2 03	1 63	255	1 49	1 27
160	1 99	1 60	270	1 45	1 24
165	1 95	1 57	285	1 41	1 21
170	1 91	1 55	300	1 37	1.19
175	1 87	1 52	315	1 34	1 17
180	1 84	1 50	330	1 31	1 15
185	1 81	1 48	345	1 29	1 14
190	1 78	1 46	360	1 26	1 13
195	1 75	1 44	420	1 19	1 08
200	1 72	1 42	500	1 13	1 05
205	1.69	1 40	600	1 08	1 03
210	1.67	1 38	700	1 05	1 01

If the belt is wrapped around two driving pulleys as in the tandem drive (page 152), the total angle of wrap may be greater than 360°, the ratio $\frac{T_1}{T_2}$ increases and the ratio of T_1 to the horsepower pull decreases toward 1; when that point is reached, all the strength of the belt is effective for moving the load and overcoming idler friction. Practically, the ratio never reaches unity, but for a wrap of 420°, the total tension is only 8 per cent more than the effective horsepower tension (see Table 5-L); this means that a belt for such a case need hardly be thicker than is required to transmit the horsepower pull.

Distribution of the Initial Tension. The tension diagrams, Fig. 5-7, illustrate the distribution of the initial tension in a conveyor. The conveyor is considered not running in cases 1, 2, 3. For this condition the tension is not uniform throughout the length of the belt. Since the belt is not a rigid member, but has some slack in it, the effect

of the initial tension will be to remove the slack and thus move the belt. However, movement of the belt is retarded by the internal resistance of the conveyor: the weight of the belt and the revolving

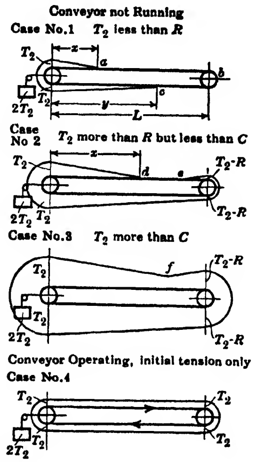


FIG. 5–7.

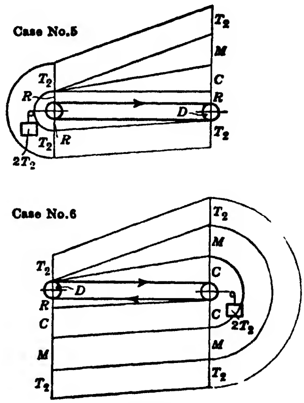
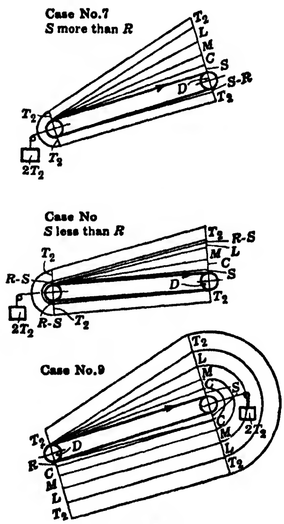


FIG. 5–8.



$2T_2$ = effective pull for initial tension

C = pull required to move unloaded belt, carrying run.

M = pull required to move material, horizontally.

L = pull required to raise material

R = pull required to move belt, return run.

S = slope pull of unloaded belt.

D = driving pulley.

FIG. 5–9.

FIGS. 5–7, 5–8, 5–9. Tension Diagrams.

parts of the idlers multiplied by the friction coefficient of the idlers. In case 1 the initial tension is assumed less than the pull required to move the return belt. It is dissipated in length x on the carrying

run and in length y on the return run, and the section of belt a–b–c is without stress. Within this section the belt could be parted without affecting the stressed portions. Case 2 shows a condition in which T_2 is greater than the internal resistance of the return run but less than the resistance of the carrying run. In this case the belt section d–e is without stress. When T_2 is greater than the internal resistance of the carrying run (case 3) all sections of the belt are stressed, the minimum stress occurring at point f. If the conveyor is running, a redistribution of the initial tension takes place. The internal resistances C and R are overcome by the effort of the driving pulley and cease to exist as a means of dissipating the initial tension, which then acts undiminished throughout the length of the belt as illustrated by case 4.

Tension in the Belt Due to Incline of Conveyor. In an inclined conveyor some parts of the belt are at a higher elevation than other parts and the belt is under stress as the result of this condition. The amount of this stress or "slope tension" is equal to the weight per foot of the belt multiplied by the vertical height of the incline in feet. The slope tension, being equal on both sides of the head pulley, does not tend to produce rotation of the belt, nor does it affect the horsepower required to run the conveyor, the additional effort to turn the upper pulley because of increased load on the bearings being neglected. With the drive located at the head and the take-up at the foot advantage can be taken of part of the slope tension to provide some of the slack-side tension required to prevent the driving pulley from slipping.

Distribution of Tension in a Belt Conveyor When Operating. The tension in a belt is the combination of the effective pull produced by the driving pulley, the initial pull introduced by the take-up, and, if the conveyor is inclined, the slope tension due to gravity. In a horizontal conveyor, Fig. 5–8 illustrates the way the tension varies. In both cases 5 and 6 the unbalanced tension at the drive pulley is equal to $M + C + R$, which is the effective pull transmitted to the belt by the driving pulley. The diagrams also show the advantage of locating the drive at the delivery end. Although the maximum belt tension $T_2 + M + C + R$ is the same in both cases, the belt in case 5 is operating under a lower average tension and the load on the take-up shaft is less.

Tension diagrams for inclined conveyors elevating material are shown in Fig. 5–9. In case 7 it is assumed that the slope tension is greater than the pull required to move the belt on the return run. The tension on the slack side of the driving pulley is $(S - R) + T_2$, the

slope tension $(S - R)$ providing some of the slack-side tension. The minimum stress occurs at the foot of the incline. In case 8 it is assumed that the slope tension is less than the pull required to move the belt on the return run. The minimum stress occurs at the slack side of the driving pulley, the same as in a horizontal conveyor. In case 9 the driving pulley is located at the foot of the incline; this causes a high average stress in the belt and a heavy pull on the take-up shaft.

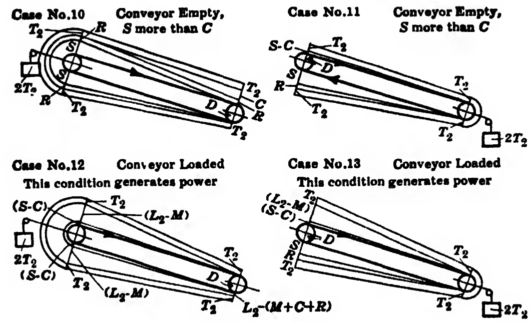
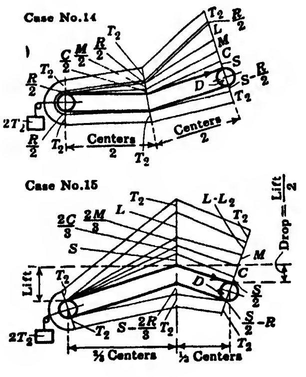


FIG. 5–10.



$2T_2$ = effective pull for initial tension.

C = pull required to move unloaded belt, carrying run.

M = pull required to move material horizontally.

L = pull required to raise material.

L_2 = pull developed by lowering material.

R = pull required to move belt, return run.

S = slope pull of unloaded belt.

D = driving pulley.

FIG. 5–11.

FIGS. 5–10 and 5–11. Tension Diagrams.

Tension diagrams for inclined conveyors lowering material are shown in Fig. 5–10. It is assumed that the slope tension is greater than the effort required to move the belt on the carrying run. Cases 10 and 11 show the tension variations for the conveyor running empty, and the unbalanced pull at the driving pulley is $C + R$. When material is conveyed the conveyors "run away" or generate power derived from a force $= L - (M + C + R)$. A combination of a horizontal and an inclined conveyor frequently used is shown in Fig. 5–11,

case 14; case 15 illustrates a combination of two inclines, one elevating and one lowering.

Where to Drive. With a single pulley driving a horizontal or an inclined conveyor elevating material the preferred location is at the delivery end. With this arrangement the average belt stress is less; this is graphically shown in Fig. 5–8, cases 5 and 6, and also in Fig. 5–9, cases 7 and 9. This means that the belt will stretch less when operating. In cases 5 and 7 the belt is not highly stressed when passing around the take-up shaft. This is an advantage in that the flexing stress in the belt is not great at that point and the pull on the take-up shaft is less. If a gravity take-up is used less counterweight is required to produce an initial pull equal to $2T_2$.

A difficulty present with driving at the foot pulley, particularly with inclined conveyors, is the disposal of the slack belt as it leaves the pulley. Usually the loading point is too close to the foot to allow any free hang of the slack belt leading from the foot pulley. The take-up cannot engage the belt on the upper or loaded side; it is often inconvenient to place it at the head because the discharge chute is there, and the alternative is to put it on the under side near the foot. This requires additional pulleys with their shafts and bearings, and the belt is under full tension passing around the take-up pulley. In spite of these drawbacks it is sometimes advisable to drive the conveyor at the foot and then take care to keep the belt at the proper tension, as in tailings stackers and similar conveyors where it is inconvenient to transmit power to the head shaft.

Types of Drives. The drive unit of a conveyor transfers the horsepower pull from the source of power to the belt. This transfer is made by friction between the driving pulley and the belt. In the effort to obtain a low ratio for $\dfrac{T_1}{T_1 - T_2}$ various driving arrangements have been developed.

The simplest type of drive consists of one cast-iron or steel pulley, connected to the source of power, and having the belt wrapped around it on an arc of about 180°. This is known as the bare pulley drive. When the carcass of a belt is determined by considerations other than tension the bare pulley drive is generally used. Its cost is least, and the belt in passing around it is bent in only one direction. The first step in decreasing the ratio of $\dfrac{T_1}{T_1 - T_2}$ from that obtained from a bare pulley drive is to add a snub pulley directly behind the driving pulley so that the arc contact of the belt with the driving pulley is increased. Another way to decrease the ratio of $\dfrac{T_1}{T_1 - T_2}$ is to increase the friction

between the belt and the driving pulley. The most common method is to cover the face of the pulley with lagging, usually with a rubber surface to contact the belt. This arrangement is known as a lagged pulley drive. A snub pulley can also be used in combination with a lagged pulley. The lagged pulley snubbed drive combines the effects of increased wrap and increased friction. Table 5–M gives the relative belt stress to transmit the same horsepower by the different arrangements of single-pulley drives.

TABLE 5–M

BELT STRESS TO TRANSMIT THE SAME HORSEPOWER WITH DIFFERENT ARRANGEMENTS OF SINGLE-PULLEY DRIVES

Belt Stress, per cent	Drive	Arc of Contact, degrees
100	Bare pulley	180
90.8	Snubbed pulley	210
81.5	Lagged pulley	180
75.0	Lagged pulley snubbed	210

A disadvantage of bare pulleys is that they wear out on the rim if the belt handles coke or similar sharp gritty material and if the belt is not carefully brushed clean. Bare tandem pulleys on coke conveyors have been known to wear out in less than a year, but when lagged pulleys were put in they did not have to be renewed. The lagging lasted eighteen months, but it is easier and cheaper to renew lagging than to replace pulleys.

Wear of Lagging. Lagging the pulleys of tandem drives sometimes leads to trouble. When the lagging wears down, the driving diameter changes, and it has happened that the second driver working against the dirty face of the belt wore smaller than the first driver, with consequent slip and a failure to drive. Unless the belt is brushed clean, fine dirt will be deposited on one of the drivers and may accumulate in thick patches. It is not easy to use a scraper (see page 246) on a lagged pulley to remove such crusts, and they may become so serious as to injure the belt or spoil the equality of belt speed on the two drivers. In some tandem drives the lagging has been removed from the pulleys for that reason, but this might not have been necessary if the belt had been brushed clean before its carrying side touched the driver.

FIG. 5–12. Worn Bolt from Lagging of Conveyor Pulley.

Another difficulty encountered in some lagged tandem drives is

that, when the lagging wears thin, the bolts which hold it project and are then likely to cut and tear the conveyor belt. Fig. 5-12 shows such a bolt, which, when the lagging wore away, projected, bent over, and then rubbed against the belt until half of the head was worn away.

The remedy for this trouble is to use pulleys with thick rims, and with a large blunt-ended drill to remove the metal from around the bolt holes so that when the bolt is drawn tight it will pull the lagging down into the countersink. Then the bolt head will be far enough below the outer surface to allow some wear of the lagging before the bolt head touches the belt (Fig. 5-13).

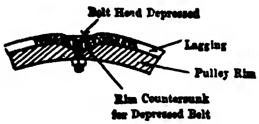


FIG. 5-13. Countersinking Rim of Pulley for Lagging Bolts.

Pulley Lagging. Belt makers supply pulley lagging made of two, three, or four plies of fabric with a "friction" back and a wearing face of high-grade rubber ⅛ or $\frac{3}{16}$ inch thick. Sometimes the rubber is held on with "tie-gum" construction. Such lagging costs more than the plain belting or scraps of belting often used to cover pulleys, but it lasts much longer and is cheaper in the end, considering the expense of repairs and renewals. The thick, strong rubber allows the heads of the fastening bolts to sink in flush; it does not flake off or glaze, but maintains a smooth surface and a strong driving grip on the belt. For solid-woven lagging see page 67.

For pulley lagging made of asbestos brake-lining see page 278.

Some rubber manufacturers will apply a layer of plastic rubber to pulleys sent to their works and vulcanize it so as to adhere very firmly to the pulley face.

Belt Must Be Kept Clean. On account of the movement between the pulley rims and the belt, caused by creep (not necessarily slip), it is very important to brush the belt well and keep it clean. It is cheaper to wear out brushes than to wear out lagging or the conveyor belt.

Multiple-Pulley Drives. From Table 5-K it is apparent that the ability of a pulley to drive a belt increases rapidly when the angle of wrap exceeds 240°. It is seldom possible to get more than 200° on a plain pulley drive or more than 255° with a snub pulley, but by using a second drive pulley engaging a reverse bend in the belt the combined angle of wrap may be made 360° or even more.

Tandem Drives. The action of tandem-pulley drive is shown in Fig. 5-14. If the pulleys A and B are of the same size and revolve at the same speed, the wrap on A being 180° and on B 240°, the fol-

lowing ratios exist when the pulleys are lagged with rubber: $\frac{T_1}{T_x} = 3.00$, $\frac{T_x}{T_2} = 4.33$. Therefore $T_1 = 3.00 \times 4.33T_2 = 13.00T_2$ (see Table 5–K). That is, for a comparatively low tension T_2 in the leaving belt it is possible to get a high driving tension T_1 in the entering belt and a high horsepower pull.

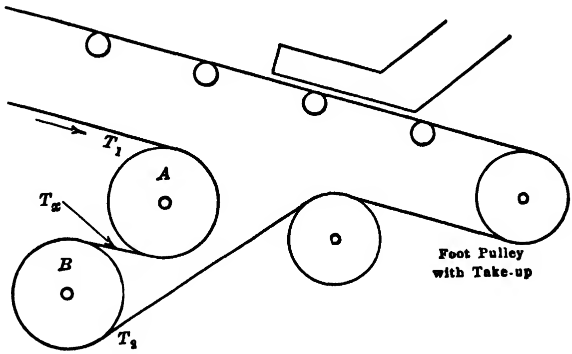


FIG. 5–14. Belt Tensions in a Tandem-pulley Drive.

Fig. 5–15 shows a tandem drive for a 60-inch 12-ply belt 2225 feet long. The belt has a wrap of about 250° on each of the two driving pulleys.

Tandem Drives on Return Run. It often happens on inclined belt conveyors or on long horizontal conveyors that to drive at the head end means costly supports and placing the machinery in a dirty or unhandy place. A tandem drive on the return run near the foot

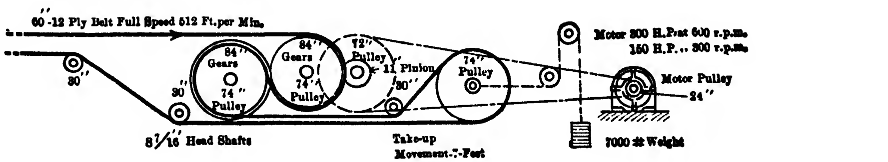


FIG. 5–15. Tandem Drive for 60-inch 12-ply Belt. (Robins Conveying Belt Company.)

puts the machinery in a cleaner place where it is more likely to receive proper attention because it is easily accessible. One drawback, not serious, is the added tension necessary to take care of the component of belt weight due to the incline (see page 148). Other points to consider are the added cost of the driving machinery and bend pulleys as compared with a drive at the head, and the general objections to

reverse bending and belt creep and slip (see page 159). Fig. 5–16 shows a geared tandem drive on the return run of a conveyor.

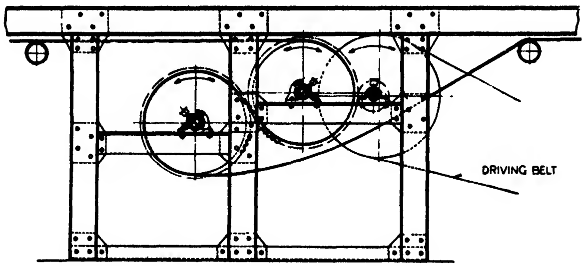


FIG. 5–16. Tandem Drive on Return Run of Conveyor.

Tandem Drive with Pulleys of Different Diameters. It is possible to gear a snub shaft to a head shaft and make the snub pulley help to drive the belt. Such a drive is shown in Fig. 5–17, which illustrates the arrangement at the head of a number of package conveyors in the Chicago Railway Post Office Terminal. This gives a driving wrap of 160° on the snub pulley and 240° on the head pulley. It is in effect a tandem drive with the speeds of the shafts in the inverse ratio of their pulley diameters. The belts are stitched canvas, 4- or 5-ply, 36 to 48 inches wide, of 32-ounce duck.

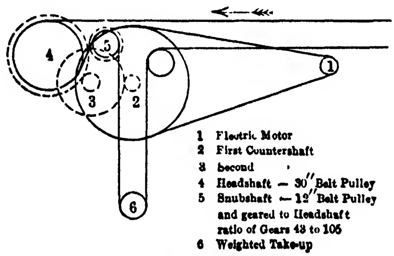


FIG. 5–17. Snub-shaft Geared to Head Shaft and Acting as Second Pulley of Tandem Drive. (Lamson Company.)

Lagging Tandem Pulleys. From Table 5–K it appears that for a combined wrap of 360° on two bare iron pulleys the ratio of $\frac{T_1}{T_2}$ is 4.80 and that for rubber-lagged pulleys the ratio is 9.02. This shows that for the same slack tension T_2 the driving tension in a conveyor belt can be nearly doubled by lagging the pulleys. For driving some long or heavily loaded belts it may be advisable to lag the pulleys to obtain a high pull with a comparatively slack belt leading from the second pulley (as the belt runs); but for others it is better to use bare pulleys and by means of take-ups, preferably weighted, make T_2 high enough to cause the bare pulleys to drive.

What this means in practice is stated in Table 5–N, which shows that, in a large conveyor belt exerting 10,000 pounds effective horsepower pull, T_1 must be increased 1300 pounds and T_2 the same amount,

if the pulleys are bare and not lagged. This represents an increase of 10 or 12 per cent in the belt tension on the loaded side; it may not be objectionable if the unit stress in the belt is not excessive, or, in other words, if the pounds pull per inch per ply is within reasonable limits (see page 169). Of course, means must be provided for giving the belt the necessary tension T_2, as it leaves the second driver, by a screw take-up or preferably a weighted take-up placed near the driving group, and not at some remote point. There is an incidental advantage in having means for adjusting the slack tension in such drives; in cold weather when the belt is stiff and perhaps frosty, the coefficient of belt contact is low and the pulleys may slip in starting up for the day. If the slack tension is increased for a short time, the pulleys will take hold, and then, after the belt has gone around a few times and is in good working order, the tension can be reduced to the normal amount.

TABLE 5-N

COMPARISON OF BELT TENSIONS IN TANDEM DRIVES

	Bare Iron Pulleys	Lagged Pulleys
Horsepower pull	10,000 lb.	10,000 lb.
Wrap in degrees .	360°	360°
T_1 (see Table 5-L)	12,600 lb.	11,300 lb.
$\frac{T_1}{T_2}$ (see Table 5-K)	4.80	9.02
T_2	2,600 lb.	1,300 lb.
Increase of T_1 .	1,300 lb.	
Increase of T_2 .	1,300 lb.	

Tandem Drives: Wear of Pulleys and Lagging. Two factors cause the wear of pulley rims and lagging referred to above, namely, **belt slip** and **belt creep.** Most persons acquainted with belts know what belt slip is; they have seen it. If the belt stands still and the pulley turns within it, the slip may be called 100 per cent of pulley travel because the travel of the belt has fallen to zero. When the pulley begins to drive the belt the slip is less, and when the drive is working properly the slip is least.

Belt creep is not so well understood because it cannot be seen. Belt creep is the *elastic* change in belt length on a pulley; it is caused by the driving effect of the pulley.

Page 372 discusses belt creep in belt elevators (see Fig. 21-1), but in tandem drives for belt conveyors the creep is much greater because

the ratio of tensions instead of being 2 or 3 to 1 may be 9 or 10 to 1. A length of conveyor belt that measures 12 inches under no tension may be stretched to 12$\frac{3}{16}$ inches as it meets the first driving pulley but when it leaves the second driving pulley the tension may be so low that the stretch is practically zero. This means that 12$\frac{3}{16}$ inches of belt has shortened $\frac{3}{16}$ inch in passing through the drive, and if the belt speed when contacting the first driving pulley is 400 feet per minute the creep is about 74 inches per minute. The wear on the pulley rim is represented by that much movement of two surfaces under severe pressure with particles of grit between them.

Seventy-four inches per minute does not seem much but running 8 hours per day and 250 days per year it is equivalent to 141 miles. It must be said, however, that the wear caused by creep is insignificant in most cases; it is only where the pull is severe and gritty material adheres to the belt that the wear on pulley rims or pulley lagging can be called objectionable, even in a tandem drive. So far as the belt is concerned the wear from creep is spread over a surface so much greater than the surface of the pulley rim that it is seldom noticeable.

Belt creep is unavoidable; it exists in all belt drives. In plain drives on bare pulleys it is least because the ratio of the tensions T_1 and T_2 is least; and as the ratio between the two tensions increases the creep increases until it reaches a maximum in lagged pulley drives. In a tandem drive there is creep on both of the driving pulleys, and if the rim speed is the same for both pulleys the creep on the first driving pulley is transferred to the second driving pulley and becomes slip and is added to the creep present on the second driving pulley.

Experiment with Tandem Drive. To test the theory of the tensions in a tandem drive, an experiment was made with a 5-inch belt on 18-inch pulleys (Fig. 5–18). Pulleys 1 and 2 made a tandem drive with a wrap of 230° on each pulley; a brake acted on shaft 3, and a tension weight was applied to shaft 4 so that A represented the tight run of a conveyor belt and B the slack run. One motor drove 1, another motor drove 2. The following table shows the current taken by each motor as the brake load was increased:

Brake, Hp.	Amperes, Motor 1	Amperes, Motor 2	Work Done by 1, per cent	Work Done by 2, per cent
3	9	3 5	72	28
6.75	19	8	70	30
8	23	9	72	28
8.12	23.5	9	72	28

To check this with theory, assume that the tension in A is 100 pounds; then, for $f = 0.25$ and a wrap of 230°, the ratio of tensions is about 2.7 (Table 5-K) and the tension on the opposite side of pulley 1, to maintain a pull of 100 pounds in A, is $\frac{100}{2.7} = 37$ pounds. To maintain a pull of 37 pounds in the short run of belt between 1 and 2, the tension on the opposite side of 2 must be $\frac{37}{2.7} = 14$ pounds. The work done by pulley 1 is measured by the difference of the belt tensions on it, or $100 - 37 = 63$, and the work of pulley 2 is measured by $37 - 14 = 23$. The total

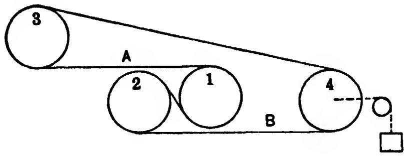


FIG. 5–18. Experimental Tandem Drive. (Link-Belt Company.)

work is $63 + 23 = 86$, and of this total, 1 does $\frac{63}{86} = 73$ per cent and 2 does $\frac{23}{86} = 27$ per cent. This agrees closely with the results of the test, but the ratio is not a constant one; with lagged pulleys and other angles of wrap, and other belt tensions the work done by the two pulleys would be in some other ratio.

In general the first pulley or main driver of a tandem does more work than the second pulley. Preferably it should work on the clean side of the belt and should be lagged. If the second pulley is not lagged, the rim may be kept clean by means of a scraper.

Tandem Drive with Two Motors. Fig. 5–19 shows a two-motor drive for an inclined coke conveyor built under the Reid patent 1,499,-319 of 1924. The first or primary motor does most of the work as has been shown above; it is usually a constant-speed synchronous motor. The secondary motor, which is of the slip-ring type, takes a smaller share of the load but its important function is to keep the belt tight between the two drivers. After the resistance of the secondary motor has been adjusted to give the proper speed to the belt that motor will transmit its power to the belt constantly at that speed. The primary motor supplies the amount of power in excess of that furnished by the secondary motor.

In this kind of drive, known as a dual drive, the two motors and their controls act as a kind of electric differential to do what is done by mechanical means in some of the one-motor geared-pulley tandem arrangements referred to later. It is necessary to make a careful study

of the electrical characteristics of the two motors in a "dual" drive.

Geared Tandem Drives. In Fig. 5–19 each of the drivers has its own motor, the size of which can be chosen to deliver to the belt the proper amount of power. In many tandem drives the two pulley shafts are geared together and are driven by one motor; it is then necessary to build the machinery so that each pulley does its proper share of the work and so that the belt does not slip on them. This means that the amount of belt paid off by the first driver should not be more or less than the amount taken in by the second driver.

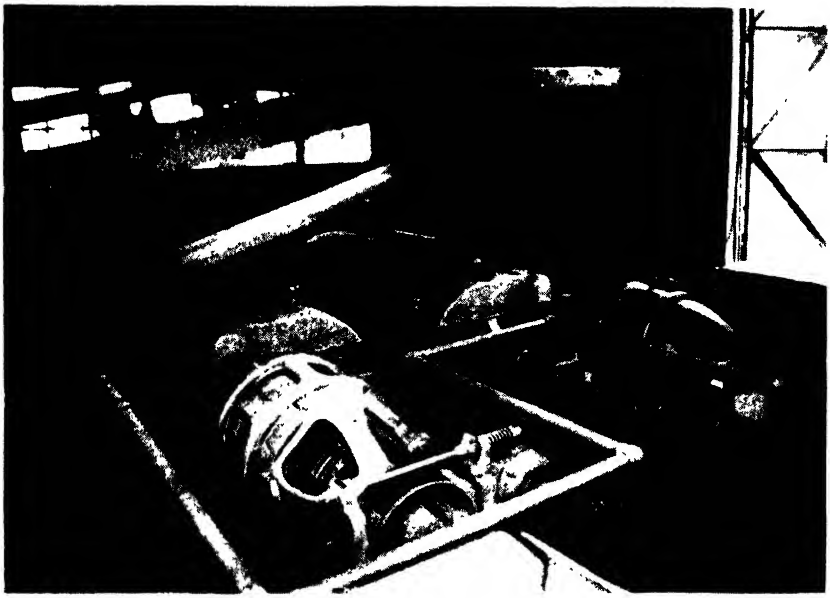


FIG. 5–19. Two-motor Tandem Drive. (Link-Belt Company.)

To get this result, the two drivers should be alike in diameter and the gears should have a 1 to 1 ratio.

This statement needs a slight correction. When the belt has a thick cover on one side and a thinner one on the other side, the outside diameter of the pulley which engages the thick cover should be slightly less than that of the pulley which bears against the thinner cover in order that the effective radius from pulley center to middle of the belt carcass shall be the same on both pulleys. This is because the belt pull is transmitted through the plies of fabric and not through the rubber covers. If the carrying side of the belt shown in Fig. 5–15 had a $\frac{1}{4}$-inch cover and the other side a $\frac{1}{8}$-inch cover,

the second driver B bearing against the $\frac{1}{8}$-inch cover should be $2(\frac{1}{4} - \frac{1}{8}) = \frac{1}{4}$ inch larger than pulley A which drives on the thicker cover. To get this difference the pulleys can be turned to the exact diameters, or one may be lagged and the other not lagged, or one may have a thicker lagging than the other.

To make sure that the first driver of a geared tandem would pull its utmost, Robins and Hersh made the second driver slightly larger than the first, obtaining patent 828,341 in 1906 on the device. With that arrangement, the belt at T_x in Fig. 5–14 would always be tight and for a given value of T_2 the value of T_1 would be a maximum. A disadvantage is that, with a given take-up, tension T_x remains fairly constant, but with a light load or no load on the belt, T_1 may fall below the value which T_x will drive and as a result the belt may slip and chatter on pulley A and perhaps on B also.

Compensation for Creep and Slip. In the normal operation of the drive shown in Fig. 5–15 the belt when on A is under higher tension and stretch than when it is on B, hence in passing from A to B it shortens. It might be possible to avoid some or all of the resulting slip on B by making the second driver smaller in diameter than the first driver, but who can say how much smaller it should be? If it be made too small, it will not take the belt as fast as the first driver gives it off, the intermediate tension T_x will drop, and A will not pull its load, or at least it will not until the tensions have been readjusted after a period of slipping and chattering.

Aside from creep and slip which cannot be calculated or measured, there are other factors also beyond calculation or measure which affect the action of a belt on a pulley. These are: (1) the actual tension in the belt, whether it be at part load, full load, or overload; (2) the condition of the belt surface, wet or dry, clean or dirty; (3) the condition of the belt itself, new or old, stiff or limber. Coefficients of friction of belts on pulleys are not fixed and certain but vary rather widely according to the three factors mentioned, as may be seen from Table 22-A, page 361.

To avoid or to overcome some of the uncertainties of geared tandem drives, other kinds of driving devices have been suggested, still others have been used successfully. One operator who has had much experience with geared tandems says that the compensating mechanical devices which he has used do not seem to be necessary. His tandems have pulleys alike in diameter except for the allowance for the thickness of the belt covers, and though the creaking noise due to creep and slip can be heard at times, the wear on belts and pulley lagging has never been serious. As has been stated on page 30, the modern

tendency in belt-conveyor practice is not to depend on auxiliary devices but to make belts so that they will do the work expected of them with the simplest possible machinery and at the least cost for maintenance and repair.

Drive with Compensating Gearing. The Hegeler and Holmes multiple-pulley drive, patent 1,176,290 (1916), is a planetary gear device which distributes power to the pulleys of a tandem in such ratios that the work done by each pulley is proportionate to the difference between the load tension and the slack tension at that pulley. If it were applied to the tandem shown in Fig. 5–18, the gearing would deliver 73 per cent of the power to 1 and 27 per cent to 2. The purpose is to allow each pulley to drive a little slower or faster momentarily, according to the tension and stretch in the belt in contact with the pulley at that time. It would thus avoid the slip due to creep, and to changes in belt length due to change of load. Pulleys geared together in a fixed speed ratio and driven by one motor cannot do this. Fig. 5–20 shows the device designed for a three-pulley drive with pulleys 1, 2, 3, with a combined belt wrap of 634° for a long inclined conveyor. A few of these compensated drives have been built for two-pulley tandems with gearing designed to deliver 73 per cent of the power to one shaft and 27 per cent to the other.

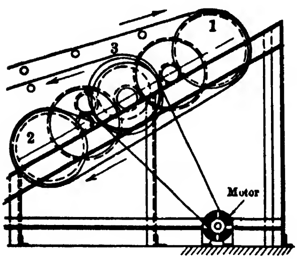


FIG. 5–20. Three-pulley Drive with Compensating Gearing.

Tandem Drive with Torque Coupling. Frank E. Smith's patent, 1,706,501 (1929), discloses a geared tandem drive with a torque coupling between the second driving pulley and the source of power. The second pulley runs faster than the first one. When the load on the second pulley exceeds the load capacity of the torque coupling, slippage occurs in the torque coupling and the excess load is taken by the first driver. The action of this drive is like that of the Robins and Hersh drive, except that slippage occurs in the torque coupling instead of between the belt and the pulleys.

Auxiliary Belt Drives. *Page's Auxiliary Drive.* This device patented in 1919 (1,313,111) is shown in Fig. 5–21. It consists of a plurality of independently driven auxiliary belts in contact with the conveyor belt and supported by the conveyor idlers so that the driving tension is applied to the conveyor belt at several points. The idea is that the maximum tension in the conveyor belt will be less and that a very long belt can safely be made lighter and cheaper. This

drive has been used only a few times. The length of a horizontal conveyor that an auxiliary belt can be expected to pull may be approximated in this way. Assume (1) that the coefficient of friction between the belts is 0.35, (2) that the journal friction for anti-friction idlers is 0.03. Then an auxiliary belt of length L will pull a section of conveyor belt, on the horizontal, equal to $\frac{0.35}{0.03}L$ or about 12 times its own length.

One limitation of this drive is that if the conveyor is loaded for only a part of its length, or if the load runs thin at intervals, the auxiliary belts under the bare places will not be so effective as under

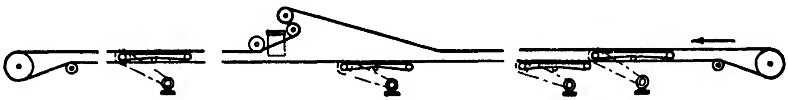


FIG. 5–21. Conveyor Belt with Auxiliary Drive Belts.

fully loaded conditions. The drive also is not suited for inclined conveyors because of the additional pull required for lifting the material.

The Hoy patent, 804,474, of 1905, shows a method of driving a conveyor belt by means of an inner auxiliary belt when it is not convenient to apply power to either of the end pulleys of the conveyor.

Driving by Auxiliary Pressure Belt. This device, known also as the "hugger" drive, is shown in the Piez patent, 1,319,-019 of 1919. It has been used instead of a geared tandem drive in perhaps two dozen conveyors. As shown in Fig. 5–22 it is a single-pulley drive with the conveyor belt pressed hard against the driving pulley by a short auxiliary belt which is kept under tension by a suspended weight. No power is applied to the pressure belt; it merely travels with the conveyor belt and exerts pressure on it to get a better grip on the driver.

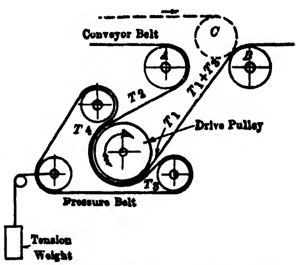


FIG. 5–22. Pressure-belt Drive.

Disregarding friction losses in the auxiliary parts, the driving effect in such a combination is the sum of the pull in the conveyor belt due to its own wrap on the driver plus the product of the tension in the pressure belt times the factor from Table 5–K which corresponds to the angle of contact between the two belts and the coefficient of friction of the one belt on the other belt. In the "hugger" drives

which have been built the two angles of wrap are approximately equal and the driving effect is that of a tandem drive with about 400° to 450° of combined wrap.

Squeeze Drives. The Dodge patent, 1,776,419 (1930), discloses the idea of increasing the pressure between the conveyor belt and the driving pulley, at the point where the conveyor belt leaves the pulley, by means of an adjustable rubber pulley pressing the conveyor belt against the driving pulley. Assume a coefficient of friction between the conveyor belt and the driving pulley of 0.35 and a pressure of 1000 pounds exerted by the rubber pulley pressing the conveyor belt against the driving pulley. Owing to this pressure a pull of 1000×0.35 or 350 pounds can be applied to the conveyor belt before it slips on the pulley at this point. The effective pull that could be developed would be $350 \times$ the factor from Table 5–K which corresponds to the angle of wrap. This arrangement permits a high value for T_1 without the disadvantage of a high value for T_2, but it may hurt the belt or the splice.

In the Fisher patent, 1,762,722 (1930), the conveyor belt is driven by being pinched between rotating driving rolls.

For "mangle-roll" drives, see page 164.

The vacuum pulley drive invented in 1931 by William S. Campbell of Glenside, Pennsylvania, proposes to obtain a high ratio for $\frac{T_1}{T_2}$ by utilizing atmospheric pressure to increase the "grip" or pressure of the belt on the driving pulley. In Fig. 5–23 a pump, driven by a separate motor, is connected so that the air in the central chamber of a driving pulley can be withdrawn and a partial vacuum created. Radially arranged valves actuated by a central cam intervene between the central chamber and the air passages that terminate in rows of slots on the face of the pulley. As each row of slots, in turn, contacts the incoming belt, the corresponding valve is opened and remains open during the period of contact. A partial vacuum is thus created over the contact area of the pulley, causing the surrounding air to press the belt against the pulley face with a force equal to the difference between the atmospheric pressure and the partial vacuum. The motor driving the belt is automatically started when sufficient vacuum has been developed within the central chamber, and stopped if the vacuum should drop below a predetermined level.

The Mathews Conveyor Company announced in 1938 that patents were pending on the drive shown in Fig. 5–24. The driving pulley and the motor are mounted on a pivoted frame and act as an automatic belt tension device.

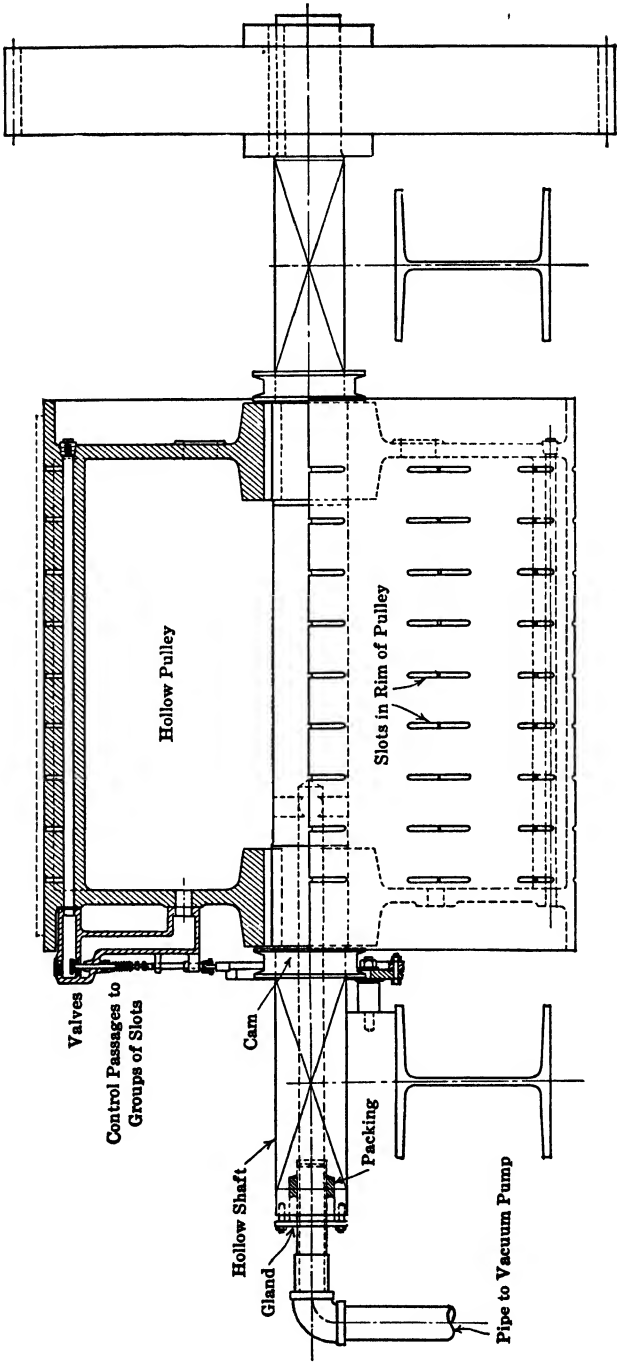


FIG. 5-23. Vacuum Pulley Drive.

Three-Pulley Drives. The three-pulley drive shown in Fig. 5–20 was patented but was never used; for one reason it would cost too much. The result it aims at can be attained more easily in another way. Suppose that we could get a combined angle of wrap of 600° on three pulleys. For a horsepower pull of 10,000 pounds, T_1 with

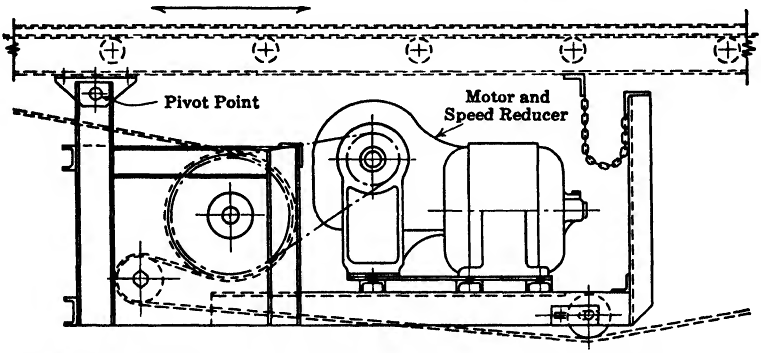


FIG. 5–24. Weight of Drive Acts as an Automatic Belt Tension Device. (Mathews Conveyor Company.)

unlagged pulleys (Table 5–L) is 10,800 pounds and T_2 = 800 pounds. If we use two pulleys with a combined wrap of 360° and lag them, we get the effective tension of 10,000 pounds with 11,300 pounds total pull in the belt. That is, by increasing the belt tension by less than 10 per cent we can make two pulleys do the work.

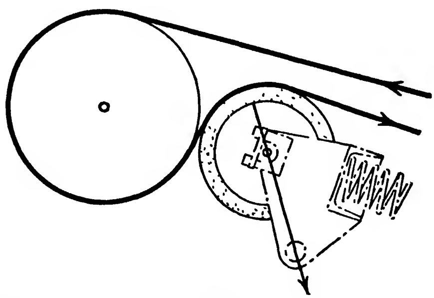


FIG. 5–25. Mangle-roll Drive. (Mining Engineering Company, Ltd.)

Driving Machinery for Conveyors in Mines. It is often difficult to fit the machinery into the small space available. In some foreign machines, the motor and its gearing are built within the head pulley—sometimes the belt is pinched between a pair of driving rolls—or is driven by the so-called mangle-roll drive with a snub pulley backed up by a spring, Fig. 5–25, or by a "two drum mangle-roll drive," Fig. 5–26.

It has been proposed (Nyborg and Higgins patent, 1,791,835 of 1931) to apply power to some of the belt idlers. An English designer has suggested * that power be applied to pairs of squeeze rolls at intervals along the empty run, the upper roll of each set being weighted and supporting also a length of the loaded belt. The lower roll would support a run of the empty belt and would be driven by a small motor.

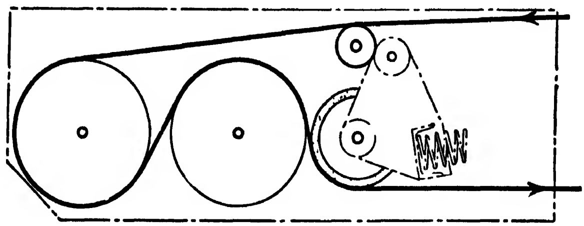


FIG. 5-26. Two-drum Mangle-roll Drive. (Mining Engineering Company, Ltd.)

It was expected that a 1-horsepower motor every 60 feet would drive the conveyor, permit the conveyor to be readily lengthened or shortened, and not require much headroom. Apparent objections are: many small parts, greater weight of machinery, injury to belt and belt splice by passing between many pinch rolls, uncertainty of driving if the belt is wet and dirty. Up to 1940 the scheme had not been tried.

* *Foerdertechnik,* 1934, No. 15.

CHAPTER 6

DESIGN OF BELT CONVEYORS

Design of Belt Conveyors. The successive steps in the design of a belt conveyor may be set down thus:

1. From known size of material select width of belt. (Table 8-E, page 216).
2. If conveyor is inclined, select a safe angle. (Table 8-D, page 213.)
3. Choose proper speed according to nature of material, size of lumps, angle of incline. (Table 8-E, page 216.)
4. From the selected speed and required capacity determine belt width. (Table 8-B, page 197.)
5. Use width as determined by 1 or 4, whichever is the larger.
6. Consider what kind of supporting idlers will be used. (Chapter 4.)
7. Calculate horsepower for belt and load, not including transmission losses. If conveyor is inclined or has a tripper, add for that. (Chapter 5.)
8. From the horsepower and belt speed calculate the effective or horsepower pull in the belt. (Chapter 5.)
9. Select the type of drive and determine the slack side tension and the maximum belt tension. (Table 5-K, page 145.)
10. Select a unit stress for the belt, and determine the number of plies and the weight of fabric. (Table 6-B, page 169.)
11. Check result of 10 with recommended maximum and minimum plies for troughed belts. (Table 6-A, page 168.)
12. Determine diameters of pulleys to be used. (Table 6-C, page 170.)
13. Select machinery between driving pulley and source of power.
14. Determine the horsepower to drive, by adding, to the power calculated from 7, the losses in the driving machinery. (Page 143.)
15. Determine diameter of drive shaft (combined bending and torsion) and other terminal shafting (bending only).
16. Consider the kind of belt, and, for rubber belt:
    (*a*) select grade of friction. (Table 6-D, page 172.)
    (*b*) select grade and thickness of covers. (Table 6-E, page 174.)
17. Size of hold-back brake, if required. Page 179.)
18. Consider location and type of take-ups and amount of tension. (Chapter 7.)
19. Consider design of loading chute and skirt boards. (Chapter 8.)
20. Consider design of discharge chute or tripper. (Chapter 9.)
21. Consider protective deck, cleaning brush, and enclosure. (Chapter 10.)

Errors or mistakes of judgment in the design of belt conveyors are not all equally serious. Some cause the belt to wear out sooner, and others can be corrected after the conveyor is in service. But there are two which are very serious; they cannot be corrected except after aggravating delay and much expense and often by humiliating makeshifts. These are a belt too narrow for the size of the material or the "peak" capacity and an inclined belt too steep. Designers should be very careful on these points.

Most of the items 1 to 21 inclusive are discussed in Chapters 4, 5, 7, 8, 9 of this book. This chapter deals particularly with items 1, 10, 11, 12, 16, which relate to the belt itself.

Selecting a Rubber Conveying Belt. Four main points to consider when selecting a belt for a specific installation are (1) width, (2) strength, (3) friction, and (4) covers.

The belt width is determined (1) by the size of the material conveyed or (2) by the tonnage handled. The speed of the belt as well as its width determine the tonnage capacity, and the belt should be wide enough to carry the volume without excessive speed. Table 8–E, page 216, gives recommendations covering the size of lumps and operating speeds for different widths of belt. For carrying capacity in tons, see Table 8-B, page 197. In selecting the width of a belt, as between the size indicated by the material and that indicated by the tonnage, whichever is greater should be chosen.

The strength of the belt depends on the number of plies and the weight of the fabric. The belt must not be too thick and stiff or it will not follow the contour of the troughing idlers, nor so thin and flexible that it will crease between the horizontal and the inclined troughing rolls. When heavy lumps are handled the belt must have sufficient body to withstand the impact of loading and handling these lumps. Table 6–A shows the recommendations for maximum and minimum plies.

There is a growing tendency on large and important belt installations to use high-grade covers (3500 to 4000 pounds) and friction (20 to 24 pounds) with skim coats between the plies. This type of belt with its thick springy cover and added rubber between the plies resists the deformation imposed by the troughing rolls to a greater extent than belts with covers of lower tensile strength. As an example, a 30-inch belt, 8-ply, 32-ounce fabric with a low tensile cover and without a skim coat troughs more easily than a 30-inch, 7-ply, 32-ounce belt having 20- to 24-pound friction with skim coats and covers $\frac{3}{16}$ and $\frac{1}{16}$ inch thick, 3500- to 4000-pound grade. It is recommended that, when working with relatively thick covers of high tensile strength and using

skim-coated carcasses, the maximum number of plies should be one less than that shown in Table 6–A.

TABLE 6–A

MAXIMUM AND MINIMUM PLIES FOR TROUGHED BELTS

Goodyear Tire and Rubber Company, 1936

Belt Width, *W*	Minimum Plies to Support Load												Maximum Plies for Troughing			
	Light Materials as Grain or Wood Chips		Fine Coal, Sand, Crushed Stone			Fine Ores, Lump Coal, Large Stone or Gravel				Coarse Ores, or Other Heavy Materials						
Inches	28 Oz.	32 Oz.	28 Oz.	32 Oz.	36 Oz	28 Oz	32 Oz.	36 Oz.	42 Oz.	32 Oz.	36 Oz	42 Oz.	28 Oz	32 Oz	36 Oz	42 Oz.
12	3	3	4	4	.	4	.		..	..			4	4		
14	3	3	4	4	.	5	4		.	.			5	4		
16	3	3	4	4	.	5	4			.	.		5	4		
18	4	4	5	4	4	6	5	.		5			6	5	4	
20	4	4	5	4	4	6	5			5			6	5	5	
24	4	4	5	4	4	6	6	5	5	6	6	.	7	6	6	6
30	4	4	5	5	5	6	6	5	5	7	6	6	9	8	7	7
36	4	4	5	5	5	7	6	6	5	8	7	6	10	9	8	8
42	4	4	6	5	5	7	7	6	6	8	7	6	11	10	9	9
48	4	4	6	5	5	8	7	7	6	9	8	7	13	12	11	11
54	...	....	7	6	6	9	8	7	6	10	9	8	14	13	12	12
60	.....	...	7	6	6	9	8	7	6	10	9	8	15	14	13	13

The maximum plies for troughing are based on standard three-pulley 20° troughing idlers (or five-pulley idlers with 15° and 30° angles). With special shallow troughing idlers somewhat thicker belts may be used if tension requires.

Within the maximum and minimum limits, the number of plies and the fabric weight are selected to give the required tensile strength. The allowable working stress for various weights of fabrics is given in Table 6–B.

The unit tensions in Table 6–B are based upon two considerations, first the tensile strength of the belt, second the severity of the flexing stress in passing over the pulleys, the larger the pulley and the thinner the belt, the less the flexing stress. The tensions stated in column 1 are recommended for general practice and where belts of average or medium grade run over pulleys of normal size as given in Table 6–C. If the belt is of good grade with 16-pound friction or better, and the pulleys are larger in diameter than normal, then the values of column 2

may be used. For instance, if Table 6–C says that a 6-ply belt of 32-ounce duck requires a 36-inch head pulley as normal, a 42-inch pulley would be larger than normal and would permit the belt to be stressed to 30 pounds per inch per ply (column 2) instead of 27 pounds (column 1).

TABLE 6–B

UNIT BELT STRESS

Fabric Weight in Ounces 42 in. wide by 36 in. long	Allowable Unit Stress in Pounds per Ply Inch, at Point of Maximum Tension			Standard Tensile Strength in Pounds per Inch Ply
	Normal Pulleys 1	Large Pulleys 2	Ideal Rating 3	
28	24	26.5	29	280
32	27	30	32.5	300
36	30	33	36	325
42 regular	38	42	46	375
42 special long staple	42	46	50	

The "ideal ratings" given in column 3 are permitted under the following considerations: (1) very large pulleys so as to reduce flexing stresses to a low point; (2) high-grade friction (at least 20 pound) and skim coats between the plies; (3) the avoidance of high starting stresses in the belt by the use of a delayed-action starting device so that the torque is not instantaneous but is spread over 20 or 30 seconds of time; (4) vulcanized splices instead of metal splice plates; (5) gravity take-ups to put a definite tension on the belt and to avoid the uncertainty of screw take-ups.

Not all belts can be used under the conditions stated above; it is certain, however, that, as those conditions are approached or approximated, belts give less trouble in operation and show longer life and lower cost per ton of material carried.

To illustrate the use of Tables 6–B and 6–C, assume an ordinary belt carrying crushed coal and stressed to 4000 pounds. A 30-inch width can be used. Of what duck, and how many plies should it be? From column 1, Table 6–B, we see that a 5-ply 28-ounce belt on normal pulleys could be stressed to $24 \times 30 \times 5 = 3600$ pounds, but this is not enough. A 28-ounce 6-ply belt on normal pulleys would be rated at 4320 pounds and would be strong enough, but might cost too

much. There are several alternatives. 1. Use a 5-ply 28-ounce belt and run it over pulleys of 30-inch, 24-inch, and 18-inch diameter for head, foot, and snub, respectively. These are larger than the normal of Table 6–C and in a good belt would qualify a rating of $26.5 \times 30 \times 5 = 3975$ pounds. 2. Use 5 plies of 32-ounce duck, which on normal pulleys is good for $27 \times 30 \times 5 = 4050$ pounds or else $30 \times 30 \times 5 = 4500$ pounds if the pulleys are 36-inch, 30-inch, and 24-inch, respectively. In this latter instance the 32-ounce 5-ply belt would deliver a pull of $\frac{4000}{30 \times 5} = 26.7$ pounds per inch per ply. This low unit tension might be highly advisable under certain conditions.

To show the trend toward heavier duck and higher unit stresses, we may mention that in 1939 a 54-inch belt of 9 plies of 48-ounce duck was made by the United States Rubber Company to be worked at 60 pounds per inch per ply.

TABLE 6–C

NORMAL PULLEY DIAMETERS FOR CONVEYOR BELTS

Goodyear Tire and Rubber Company, 1936

No of Plies	28 Oz.				32 Oz.				36 and 42 Oz.			
	Tandem Drive	Head and Tripper	Tail and Take-up	Low Tension Snub	Tandem Drive	Head and Tripper	Tail and Take-up	Low Tension Snub	Tandem Drive	Head and Tripper	Tail and Take-up	Low Tension Snub
3	18	15	12	10	24	18	15	12				
4	24	20	18	12	30	24	20	15				
5	30	24	20	15	36	30	24	18	42	36	30	24
6	36	30	24	18	42	36	30	24	48	42	36	24
7	42	36	30	24	48	42	36	24	54	48	36	30
8	48	42	30	24	54	48	36	30	60	54	42	36
9	54	48	36	30	60	54	42	36	72	60	48	36
10	60	48	42	30	72	60	48	36	72	60	54	42
11	60	54	48	36	72	60	48	42	84	72	60	48
12	72	60	48	36	84	72	54	42	84	72	60	48
13	72	60	54	42	84	72	60	48	96	84	72	54
14	84	72	60	42	96	84	72	48	96	84	72	54
15	84	72	60	48	96	84	72	54	96	96	84	60

Snub and bend pulleys placed at points where the belt tension is high should be at least as large as the "tail and take-up" classifications above.

Grades of Friction. Table 3–B, page 55, shows that American belts are classified in three grades of friction and four grades of cover. British belts are shown in Table 3–C. To select a belt with a grade of friction proper for the work requires a consideration of (1) the severity of the flexing strains; (2) the frequency of those strains. The severity

varies directly with the tension in the belt and the belt thickness and inversely as the diameters of the pulleys; the frequency increases directly as the speed of the belt and decreases as the length of the conveyor.

Table 6–D adapted from the Goodyear handbook gives in column 1 the time it takes the belt to go once round the conveyor; from columns 3, 4, 5, and 6 it will be seen that, as the time increases, a lower grade of friction may be used, other things being equal, because the frequency of the flexing decreases. The severity of the flexing becomes less if the pulleys are larger than what is called normal in Table 6–C, and in column 2 of Table 6–D normal is called 100 per cent. Comparing the values given in columns 3, 4, 5, and 6 shows that, for given values in columns 1 and 2, the higher the belt tensions, the higher should be the grade of friction in the belt.

Where no friction grade is stated in the table, some judgment should be exercised. If the hours are long and the service severe a belt with 20-pound friction will be indicated, and to insure a long life under poor operating conditions it may be advisable to have skim coats between the plies. However, if the belt works only a few hours a week a belt with a lower grade of friction may do the work acceptably. The values on the table are for belts in regular service, 8 or 10 hours a day, 5 or 6 days a week, all year round.

Skim Coats. The practice of adding a skim coat of rubber, about 0.005 to 0.007 inch in thickness between the frictioned plies of the carcass, is increasing for belts that operate under unfavorable flexing conditions and also for belts in large and important installations. Friction failure occurs not from one severe quick strain, but from small strains repeated many times. It is a fatigue failure. The skim coat does not increase the adhesion between the plies, but the additional rubber enables the friction to resist the normally small flexing strains for a longer period.

Table 6–D was first published in 1930. It is a rational guide to show the quality of friction required, and it can be followed with confidence in most cases.

Lateral Flexing. The table does not take into account lateral flexing, that is, the hinge action in troughing the belt. In conveyors up to 300 to 500-foot centers this lateral flexing does not appear to determine the life of the belt. However, as the length of a conveyor increases, the frequency of flexing around the pulleys decreases, and lateral flexing assumes greater importance. An illustration of the limitation of the table in this respect is furnished by a 48-inch, 8-ply, 32-ounce belt that needed replacement. The cover was worn through,

TABLE 6–D

GRADES OF FRICTION FOR VARIOUS CONDITIONS OF FLEXING

Time for One Revolution of the Belt around the Conveyor in Minutes	Pulley Diameters as Percentages of Normal Diameters from Table 6–C Normal is 100%	Actual Unit Stress in Belt as per cent of Rating from Table 6–B			
		50	75	100	125
		Grade of Friction, stated as pounds			
1	2	3	4	5	6
0.2	75				
"	100	20			
"	125	16	20		
"	150	12	16	20	20
0.4	75	20			
"	100	16			
"	125	12	20		
"	150	12	16	20	20
0.6	75	20			
"	100	16	20		
"	125	12	16	20	
"	150	12	12	16	20
0.8	75	20			
"	100	12	20		
"	125	12	16	20	
"	150	12	12	12	20
1.0	75	20			
"	100	12	20		
"	125	12	16	16	
"	150	12	12	12	16
1.5	75	16			
"	100	12	16		
"	125	12	16	16	
"	150	12	12	12	16
2.0	75	16			
"	100	12	16	16	
"	125	12	16	16	20
"	150	12	12	12	16
3.0	75	12			
"	100	12	16	16	
"	125	12	12	12	16
"	150	12	12	12	12
4.0 or more	75	12			
"	100	12	16	16	
"	125	12	12	12	16
"	150	12	12	12	12

and the carcass showed incipient ply separation along a path directly over the junction of the horizontal and inclined rolls of the troughing idler. The unit tensile stress in the 8-ply belt was low, so low that a 7-ply belt would safely withstand the tension. A 7-ply replacement belt was then installed but it failed after carrying only about one-half of the tonnage that was handled on the original 8-ply belt. The failure was due to ply separation along the same path as was indicated in the first belt. There was no indication of "pinching" of the belt between the ends of the horizontal and inclined rolls, as might be expected with a very thin belt, but rather the failure was typical friction fatigue. It is easy to understand that a 7-ply belt would flatten between idlers to a greater degree than an 8-ply belt and that the lateral flexing would be more severe. The table would neither have indicated this failure nor provided a recommendation that would guard against it. Only a limited number of observations of this nature have been made, and quantitative values for lateral flexing are not, as yet, available. It appears likely that the long belts of the future, having good-sized pulleys, will have their friction failures due to idler hinge action as well as to flexing around the terminal pulleys.

Protecting Covers. The four grades of covers shown in Table 3–B, page 55, vary in their tensile strength from 800 pounds per square inch for the lowest grade to 3500 pounds for the highest. The low-tensile rubbers are harder and less resilient, contain less rubber and more mineral substance, and are cheaper. The better grades are softer and tougher, and contain more strong rubber; they receive more care in manufacture and cost more. Since the purpose of the cover is to protect the fabric carcass of the belt from injury, and do it to an *economical degree,* the kind of cover and its thickness should be chosen with regard to the principles stated on page 42.

Wear on the cover depends: (1) on the severity of the cutting and abrasive action of the material loaded on the belt and carried by it; (2) on the frequency with which any place on the belt is acted upon by cutting and abrasion. The severity depends on the nature of the material and on the size and weight of the lumps or large pieces in it. The frequency increases directly with the speed of the belt and decreases as the length of the conveyor increases.

Table 6–E, adapted from the Goodyear handbook, gives in column 1 the time, in minutes, that it takes the belt to go once round the conveyor. Column 2 shows the four grades of covers given in Table 3–B, page 55, and opposite to them the thickness of the top cover in thirty-seconds of an inch recommended for the various kinds of service indicated at the head of the table.

TABLE 6-E

QUALITY AND THICKNESS OF TOP COVERS

Ordinary speeds. Average loading conditions. Quality as tensile strength, pounds per square inch. Thickness in thirty-seconds of an inch

Time for One Revolution of the Belt, minutes	Quality of Cover, tensile, pounds per square inch	Materials Not Abrasive				Abrasive Materials				Heavy Abrasive Materials				Very Abrasive, Heavy and Sharp			
		Sizes of Pieces up to (inches)				Sizes of Pieces up to (inches)				Sizes of Pieces up to (inches)				Sizes of Pieces up to (inches)			
		$\frac{1}{4}$	$1\frac{1}{2}$	5	6+	$\frac{1}{4}$	$1\frac{1}{2}$	5	6+	$\frac{1}{4}$	$1\frac{1}{2}$	5	6+	$\frac{1}{4}$	$1\frac{1}{2}$	5	6+
0 2	800	6	12	...	...	12											
"	1400	4	8	12	..	9	...	...	...	12							
"	2500	3	6	10	12	6	12	...	..	10	.	...	..	12			
"	3500	2	4	8	10	4	8	12	...	7	12	12	12	10	12	12	12
0.4	800	4	6	12		7	12	..	..	12							
"	1400	3	4	8	12	5	9	..	.	8		.	.	12			
"	2500	2	3	6	8	3	6	12	..	6	10		...	7	12		
"	3500	1	3	4	6	2	4	8	12	4	8	12	12	5	5	12	12
0 6	800	3	5	9	12	10	9			8		..		12			
"	1400	2	4	6	10	6	6	12	.	6	12	..		8	12		
"	2500	2	3	4	6	2	4	8	12	4	7	12	..	6	10		
"	3500	2	3	4	6	2	4	6	8	3	5	8	12	4	7	12	12
0 8	800	2	4	6	10	4	7	12	.	6	12	...		8			
"	1400	2	3	4	7	3	5	9	12	4	8	.		6	12		
"	2500	2	3	4	6	2	4	6	9	2	5	10	..	4	7	12	
"	3500	2	3	4	6	2	4	5	6	2	4	7	12	3	5	10	12
1.0	800	2	3	5	8	3	6	10	12	5	9	..	..	6			
"	1400	2	3	4	6	2	4	7	10	3	6	12		4	10		
"	2500	2	3	4	6	2	4	5	7	2	4	8	12	3	6	12	
"	3500	2	3	4	6	2	4	5	3	2	4	6	8	2	4	8	12
1.5	800	2	3	4	6	2	4	7	12	6	6	12	...	6			
"	1400	2	3	4	6	2	4	5	7	2	4	8	12	4	8	12	
"	2500	2	3	4	6	2	4	5	6	2	4	6	8	2	4	8	12
"	3500	2	3	4	6	2	4	5	6	2	4	6	7	2	4	6	8
2.0	800	2	3	4	6	2	4	6	8	3	6	10	...	4			
"	1400	2	3	4	6	2	4	5	6	2	4	7	10	3	6	10	
"	2500	2	3	4	6	2	4	5	6	2	4	6	7	2	4	6	12
"	3500	2	3	4	6	2	4	5	6	2	4	5	6	2	4	6	8
3.0 and over	800	2	3	4	6	2	4	5	6	3	5	6	.	4			
	1400	2	3	4	6	2	4	5	6	2	4	6	8	3	6	8	
	2500	2	3	4	6	2	4	5	6	2	4	6	7	2	4	6	8
	3500	2	3	4	6	2	4	5	6	2	4	5	6	2	4	6	8

The column heading "not very abrasive" means steam sizes of anthracite coal, prepared foundry sand, clean gravel, pebbles, wood chips, flue dust, loose cement, bone char. As "abrasive" materials may be listed sharp sand, rock salt, run-of-mine coal, soda-ash, soft ores, excavated earth. "Very abrasive" means most ores of iron, copper, zinc, and lead; limestone; shale; slag; foundry refuse; and coke. The last column heading includes such materials as trap rock, quartz, basalt, glass cullet, iron pyrites, and coke.

Lump material with sharp edges is much more abrasive when wet than when dry.

Table 6-E must be used with some judgment and with some knowledge of what has been done in practice; Chapter 14 may give some hints on this. If the load is much heavier than usual, or the conditions of loading unavoidably bad, and if the belt works three shifts a day instead of one, the belt may be out of the range of the table altogether. For instance, "mucker" belts used to carry blasted rock away from tunneling machines may have covers ½ inch thick, yet they wear out quickly. On the other hand, the throwing belts shown in Fig. 13-3 work under such great load and terrific speed that the work is not *conveying* in the usual sense. Good covers on these belts do not pay, cheap covers are useless, and practice has come to tough strong belts with no rubber at all, but with steel plates riveted on for some protection. In regular conveying work some belts may be intended for temporary use only or circumstances may be such that it will not pay to spend money on belt covers where a friction surface belt may be good enough.

Cover on the pulley side of the belt in ordinary conveyor service is $1/32$ inch, but a thickness of $1/16$ inch may be justified if the work is hard and if dirt clings to the driving pulley. Elevator belts in wet service often have pulley side covers heavier than $1/16$ inch; see Chapter 20.

Abrasion Due to Troughing. Table 6-E shows the recommended thickness and quality of rubber covers for protection, considering both the severity and the frequency of the abrasive action. It has somewhat the same limitation as Table 6-D. The wear on the cover at the loading point only is considered, but wear also occurs as the troughed belt flattens between the idlers and is then forced back into the full troughed shape. At first thought the abrasion due to this "make and remake" of the load seems so slight that it can safely be neglected. It is true that the abrasion from each squeeze is small but the squeezes occur very often (see page 121). The relative importance of this abrasion increases as the conveyor becomes longer and

the frequency of the wear at the loading point becomes less. Illustrating this point is the record of two belts of the same specification, one used on a 779-foot-center conveyor, the other on a 1512-foot-center unit. These conveyors, operated at the same speed, were in tandem and handled the same material. The loading condition for each was excellent. When it was necessary to replace these belts, because their covers had worn through, the records showed that each had handled about the same tonnage. If wear occurred only at the loading point the belt cover on the long conveyor should have carried twice the tonnage because it passed under the loading point only one-half as many times. The recommendations in the table were developed to cover the great number of average installations. However, it appears, from some observations, that the cover life on the very long conveyors of tomorrow, with their well-designed loading chutes, may be limited by the abrasion from the idler remake of the load as well as from the wear at the loading point.

Size of Pulleys. In the $\frac{T_1}{T_2}$ formula on page 144 the only variables which determine the tractive force are the angle of wrap and the coefficient of friction; the diameter of the pulley does not enter into the calculation. Some have thought that a large pulley gets a better grip on a belt than a smaller one, but within the limits of good practice that is not true.

Haddock's experiments (*Transactions, A.S.M.E.*, 1908) with a 12-inch 4-ply rubber belt showed no variation in tractive force whether it was driven by a 42-inch pulley or a 20-inch pulley, but on a 12-inch pulley the traction dropped 20 per cent. That was because the 4-ply belt was too stiff to bend to the 12-inch diameter without loss of contact pressure. These experiments established the rule that the diameter of the drive pulley should be at least five times the number of plies in the belt. When a pulley acts as a guide or a deflector, as at a foot shaft or a snub shaft, it is considered proper to make the diameter three or four times the number of plies, but, in any case, the larger the better.

The objection to making the ratio larger than 5 to 1 for drivers is that the pulleys take up more room and cost more, and more gearing or larger gearing is required because the larger pulley makes fewer turns for the same conveyor speed. On the other hand, a diameter larger than that given by the 5-to-1 ratio stresses the friction between the plies less, makes it less likely to crack when old and hence postpones the day when the belt fails by separation of the plies.

An incidental advantage of a large head pulley is that when it is

used with a snub pulley the angle of belt wrap can be made large without having the snub pulley too small.

When the ratio for a driver is less than 4 to 1 the belt tension must be increased to make the belt hug the pulley tight enough to drive. This in itself is a disadvantage, but the greater harm is the excessive stretch in the friction layers between the plies and the greater tendency for the plies to come apart when the friction gets old.

The ratio of pulley diameter to number of plies of belt in the discussion above is for 28-ounce fabric. For heavier fabric the ratio should be greater; see Table 6-C.

Pulley Rims. Drive pulleys, foot pulleys, and snub pulleys should be specified as "double-belt" pulleys generally with a "crown" on the pulley face of at least 1/8 inch per foot of face. A heavy crown will keep the belt centered even if the shaft on which the pulley is mounted should get out of level or out of square with the belt; hence some designers specify 3/16- or 1/4-inch crown per foot of face for drive pulleys where the belt might tend to get out of line. Pulleys wider than 24 or 30 inches should have double arms. The faces should be 2 inches wider than the belts up to 42 inches, 3 inches more for wider belts. This excess of width permits the belt to run out of center for a few inches without requiring it to be "trained" back into position by adjusting the troughing idlers or forcing it back by means of edge rolls. It is quite necessary where the conveyor head or foot is supported on a frame which may settle or get out of line or where the conveyor runway is not permanently aligned. A few inches of excess width on the troughing and return idlers is also worth having under such conditions. It is better to incur that expense than to ruin a belt by the use of side-guide idlers or to damage the pulley side by skewing the troughing rolls to an excessive angle. For very heavy pulls, the rims of drive pulleys should have inside flanges for reinforcement.

Straight-face pulleys have been used to drive heavily stressed belts. They eliminate a secondary stress in the belt that is often overlooked, the stress caused by the crown of the pulley. The diameter of a crowned pulley for a 48-inch belt is 1/2 inch larger at the center than at the edges, and the difference in diameters stretches the central portion of the belt more than the edges. Usually the maximum tension in a belt is calculated without including this tension, but by eliminating it highly stressed belts are helped. It has been found that the lagging on straight-face pulleys lasts longer than on crowned pulleys. On one installation crowned pulleys were transformed to straight-face pulleys by means of special beveled lagging. This lagging not only wears longer, but also prevents slippage when the belts become wet.

Snub Pulleys. If a conveyor belt makes a wrap of 180° on an iron pulley the tension T_2 in the empty belt must be $\frac{1}{2.19} = 0.456$ of that on the loaded side (see Table 5-K). If the useful pull or horsepower pull is 2000 pounds, the total tension T_1 on the tight or loaded side must be $2000 \times 1.84 = 3680$ pounds (see Table 5-L, page 146), and therefore the pull on the empty side is $3680 \times 0.456 = 1680$ pounds. But if a snub pulley (see page 151) is used to increase the angle of wrap to 240°, the total belt tension T_1 is, for the same work, 3100 pounds, and the pull on the empty side T_2 is 1100 pounds, a reduction of nearly 600 pounds in the belt tension, equivalent to 1 ply of a 24-inch belt. This is the advantage of using a snub pulley; to offset it, there is the expense of the pulley and the reverse bending of the belt, also the tendency for sharp particles adhering to the belt to be forced into the cover by contact with the snub pulley.

For most horizontal conveyors handling coal a snub pulley is not required, and in other conveyors handling heavier materials it may be better to work the belt at a tension higher than normal rather than install a snub pulley.

On conveyors handling materials which stick to the belt, a snub pulley is likely to receive a crust or layer which may build up so as to be objectionable. A steel scraper bearing against the face of the pulley will prevent it (see Chapter 10).

Deflector pulleys around which the conveyor pull is transmitted should be as large as drive pulleys, that is, 5-inch diameter per ply of belt. Other pulleys can be 3 or 4 inches per ply of belt. But see also Table 6-C, page 170. An instance where the total pull is transmitted around deflector pulleys is shown in Fig. 6-1, representing an inclined coke conveyor driven at the foot B. Instead of making the head and foot pulleys A and B 36 inches and the deflector and take-up pulleys C and D 24 inches it would have been better to make all 36 inches or at least all 30 inches because the belt was under practically the same tension in passing around all of them. The deflector E was 16 inches in diameter, but 24 inches would have been better for a similar reason. The belt in this case was 36 inches wide, 5 plies of 28-ounce duck, although the work required only 2 plies to take care of the maxi-

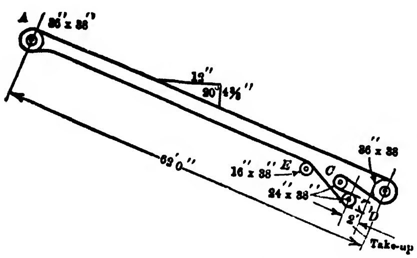


FIG. 6-1. Conveyor with Deflector Pulleys too Small in Diameter.

mum tension T_1. It lasted fourteen months and carried 240,000 tons of crushed coke at a cost of one-ninth of a cent per ton, a fairly good record; but if the pulleys had been larger it might have lasted longer than it did. Its fourteen months of service were not enough to make it die of old age.

Welded steel pulleys have been used on some important conveyors. In large sizes, they are lighter and cheaper than cast-iron pulleys.

Automatic Hold-back Brakes. When a belt conveyor lifts material and the power required for lifting exceeds the power to operate the empty belt and to move the material horizontally it is a wise precaution to install an automatic hold-back brake. If such a conveyor is loaded and the current supplying the motor is interrupted, intentionally or otherwise, the belt loses its speed, stops, and then reverses its direction. The weight of material on the incline overcomes the frictional losses in the conveyor, and the material is carried downhill and backward into the loading chute. This causes a jam of material, skirt plates, and belt, and the result is sometimes a torn belt. An automatic hold-back brake, permitting the driving machinery to turn in only one direction, will prevent the reversal of the belt when the power is cut off. Differential band, ratchet and pawl, and solenoid brakes are used for this purpose. The size of the brake is determined by the difference in power between lifting the material and operating the loaded conveyor horizontally.

Capacity of Brake. If a loaded belt drifts backward only a few feet it may jam material at a loading chute or under a foot pulley and damage the belt. In order to prevent such an accident the brake should have a torque capacity higher than that theoretically required so that the conveyor is stopped promptly and with certainty. On a long, heavily loaded conveyor 50 per cent excess of torque capacity may not be too much.

A brake may be strong enough to stop and hold the machinery but it may not prevent the loaded belt from slipping backward on the head pulley and running down hill if the slack-side tension in the conveyor belt falls too low, or if the belt is wet. It is important in inclined conveyors to place the weighted take-up close to the driving machinery so that slack due to belt stretch is removed promptly and the suspended weight acts with full effect.

It may be thought that an amount of slack-side tension enough to drive an inclined conveyor should be able to prevent it from running backward. But if there is slack enough in the return belt or in the driving machinery or in the brake apparatus to allow the belt to *start* down hill for a foot or two, the momentum of the moving mass may

keep it moving in spite of the brake, or else cause the belt to slip around the braked head pulley and run backward.

Design of a Specimen Belt Conveyor. Problem: to carry 2000 tons of limestone in 10 hours from a crusher up an incline to a bin. Crushed rock ranges in size from fines to 15 inch; it weighs 100 pounds per cubic foot. Crusher is fed by dump-cars and is rated by the maker at 250 tons per hour.

A preliminary layout gave 225 feet centers for a conveyor inclined about 18° and a vertical lift of 70 feet. Because of the intermittent feed to the crusher and the momentary increase in crusher output when the rock ran small and did not require much reduction in size, it was thought that a capacity at the rate of 330 tons per hour should be provided for instead of the 250 tons based on average hourly service.

Calculations. We will consider items 1 to 21 inclusive listed at the opening of this chapter.

1. *Belt width.* According to Table 8-E, page 216, the belt should be 36 inches wide to handle a mixture containing a good percentage of 15-inch pieces.

2. *Angle of Incline.* Table 8-D, page 213, says that 18° is a proper angle for the material, especially as it will contain some fines.

3 and 4. *Speed of Belt.* Table 8-B, page 197, shows that a 36-inch belt will carry 240 tons per hour of 100-pound material at 100 feet per minute with a moderate (20°) surcharge of load. To get 330 tons per hour with some margin of safety we will run the belt at 150 feet per minute. At this low speed, there should be no trouble in loading on the 18° slope or with picking up the large pieces at the loading point.

5. It was decided to use a 36-inch belt at 150 feet per minute and call the capacity 350 tons per hour.

6. *Supporting Idlers.* On account of carrying heavy material in large pieces it was thought proper to use idlers with 6-inch steel rolls of the style shown in Table 4-A rather than idlers with smaller rolls, higher stands, and lighter construction.

7. *Horsepower.* We estimate this for the empty conveyor from Table 5-E as follows. For a 36-inch conveyor 225 feet centers at 100 feet per minute, the horsepower is between 1.27 and 1.45. Call it 1.36; then the horsepower at 150 feet per minute is 2.1. Interpolating in Table 5-F, the horsepower to convey material horizontally is about 1.14 for 100 tons per hour on 225 feet centers; for 350 tons it is about 4.0 horsepower. To lift the material we refer to Table 5-J; 350 tons per hour lifted 70 feet calls for 24.7 horsepower. The total is $2.1 + 4.0 + 24.7 = 30.8$ horsepower.

8. *Effective or horsepower pull* for 30.8 horsepower at 150 feet per minute belt speed is $30.8 \times \frac{33000}{150} = 6776$ pounds.

9. *Belt Tensions.* If we assume that the belt will be driven by a lagged pulley snubbed to give a wrap of 210°, the value of T_1 corresponding to a horsepower pull of 6776 pounds will be, from Table 5-L, $6776 \times 1.38 = 9350$ pounds, and T_2 will be $9350 - 6776 = 2574$ pounds.

10. *Unit Stress in Belt.* From Table 6-A we see that a 36-inch belt handling coarse, heavy stone, should not have less than 6 plies of 32-ounce or 36-ounce duck or 5 plies of 42-ounce fabric. The unit tensions in such belts to transmit 9350 pounds pull would be: 32.4 pounds per inch per ply for 8-ply, 37.1 pounds for 7-ply, and 43.3 pounds for 6-ply. Nine plies of 32-ounce duck or 8 plies of 36-ounce could be used with large pulleys without exceeding the stresses given in Table 6-B, but it seemed advisable, considering costs, to use 7 plies of 42-ounce duck at a unit stress of 37.1 pounds. This would be satisfactory with "normal" pulleys and would give a fair margin below the 42 pounds per inch per ply allowed for 42-ounce duck when run on large pulleys. The 42-ounce duck would resist bumps and shocks better than the lighter ducks, and 7 plies of it would cost only about ½ of 1 per cent more than 8 plies of 36-ounce and about 5 per cent less than 9 plies of 32-ounce duck.

11. Seven plies of 42-ounce duck is between the minimum and maximum limits given in Table 6-A.

12. From Table 6-C we take a 48-inch head pulley as the normal size for the conveyor under consideration. It will make 12 revolutions per minute for 150 feet belt speed.

13-14. If the motor will make 1800 revolutions per minute and is direct-coupled to a speed reducer with two gear reductions, we can assume a power loss of about 6 per cent. In the transmission to the head shaft by cut gears or finished steel roller chain there is another loss of, say, 5 per cent—a total loss of about 11 per cent. The output of the motor should be $30.8 \times 1.11 = 34$ horsepower. A 40-horsepower motor will give margin enough for starting under load and will provide for the contingency that the voltage of the current at the motor may drop slightly at times.

15. This can be determined from tables published in hand books and trade catalogs.

16*a*. *Grade of Belt.* Refer to Table 6-D. The belt goes once round the conveyor in about 3 minutes (column 1); it runs on pulleys of normal diameter (100 per cent in column 2); the unit stress is 37 pounds, which is about 90 per cent of its rating as given in Table 6-B. Call this percentage 100; then from column 5 it appears that a 16-pound friction would be proper for the work. It is probable, however, that a 20-pound friction would give the belt a longer life.

16*b*. *Thickness of Cover.* Refer to Table 6-E. The belt goes once round the conveyor in about 3 minutes. Opposite that figure in the first column look in the column headed "heavy abrasive materials" under "size of pieces, 6 inches plus." The recommendation is a 1400-pound cover ¼ inch thick, a 2500-pound cover $\frac{7}{32}$ inch thick, or a 3500-pound cover $\frac{3}{16}$ inch thick. In choosing the last-named alternative, namely, the highest-grade cover $\frac{3}{16}$ inch thick, it was considered that the crusher plant and the belt conveyor would be operated for a dozen years or more and that probably, at the rate of five or six days a week, a loading chute could be installed to give the stone some velocity in the direction of belt travel and that a good cover would have a fair chance to show a long life. Another consideration is that the $\frac{3}{16}$-inch

cover rated at 3500 pounds will probably cost no more than the others. On account of the jagged shape of the large pieces it was thought desirable to have a breaker strip molded within the thickness of the 3/16-inch cover. A bottom cover 1/16 inch thick was decided on because the belt at the foot of the conveyor would probably work in an atmosphere full of dust.

17. If the conveyor should be stopped while loaded, its tendency to reverse its direction or run backward would be measured by the fall of 350 tons per hour, if fully loaded, less the friction losses in the conveyor. In terms of horsepower it would be (see page 180): $24.7 - 2.1 - 4.0 = 18.6$ horsepower. This is equivalent to a pull of $18.6 \times \dfrac{33000}{150}$ pounds at the rim of the head pulley or a torque of 8200 foot-pounds. Any mechanical or electrical brake used at the head shaft to prevent the conveyor from reversing its direction must be strong enough to develop that amount of torque. See also page 179.

18. *Take-up Tension.* For the proper operation of this conveyor at the assumed capacity of 350 tons per hour, the slack-side tension in the belt T_2 should be 2600 pounds; see page 180; of this a part will be supplied by the downhill component of the return belt and the rest of it must be added by take-up weight at the foot of the conveyor. The belt weighs about 12½ pounds per foot; the pull at the under side of the head pulley due to it would be $70 \times 12\frac{1}{2} = 875$ pounds. Part of this would be lost in idler friction but about 800 pounds of it is effective. The take-up must therefore add $2600 - 800 = 1800$ pounds to the lower run of the belt, and of course the same amount to the up-run. A gravity take-up with 3600 pounds of weight is required at the foot shaft.

19, 20, 21. Refer to the chapters mentioned.

If, in the example discussed above, the stone had been crushed to ballast size in the quarry and not at the top of the bin, it may seem that the belt might have been narrower than 36 inch; a 30-inch or even a 24-inch belt is wide enough to carry more than 300 tons per hour of crushed stone at moderate speed. Calculations like those for item 7 show that there would be a saving of 1 or 2 horsepower in using a 24-inch or a 30-inch belt, but the 24.7 horsepower for the lift would remain unchanged. Allowing a net effective tension of 8700 pounds, a 24-inch belt would have to be 9 plies thick to keep the unit stress under 40 pounds, and from Table 6-A that is too thick for troughing. Similarly, a 30-inch belt would require 8 plies at 36 pounds unit stress, and, moreover, if the belt had a 3/16-inch cover, 8 plies would probably not trough properly. To sum up, a 36-inch belt is recommended not only because of the large stone, but also because a narrower belt transmitting the same pull would require more plies of duck and would become too thick to lie down properly on the troughing idlers.

Drive of a Long Horizontal Conveyor. Assume a stone conveyor horizontal instead of inclined, but requiring 30.8 horsepower, the same as for the inclined conveyor discussed above. It would be about 1420 feet long, it could be driven in the same manner as the inclined conveyor, and it would require about 2900 feet of 36-inch belt with 7 plies of 42-ounce duck, about 6 times the length of the belt on the inclined conveyor. The cost of the belt on a long conveyor becomes an important item, and the design of the conveyor and its drive should receive careful attention. How can the cost of the belt be reduced? The first thought is to reduce the tension in the belt by increasing the wrap on the single-drive pulley from the assumed 210° to a possible maximum of 240°. Calculations show that this would reduce the belt pull by not more than 6 per cent, which is not enough to permit the use of a 6-ply belt. But if we use a two-pulley tandem drive, the angle of wrap can be made 420° and the belt tension will be 7318 pounds instead of 9350 pounds, a reduction of about 20 per cent. We could use 6 plies of 42-ounce duck at 34 pounds, or 7 plies of 36-ounce at 29 pounds, or 8 plies of 32-ounce at 25.4 pounds per inch per ply. These three alternative specifications for a 36-inch belt handling coarse, heavy stone are satisfactory for troughing (see Table 6-A), and the unit tensions are all lower than those which according to Table 6-B are allowable on pulleys of normal diameter; hence there would be no need to use pulleys larger in diameter than those given in Table 6-C. If we decide to use a tandem drive for this 1420-foot horizontal conveyor instead of a lagged single-pulley drive we save the price of one ply of 42-ounce duck which in 2900 feet of 36-inch belt is worth more than $2000. If we choose one of the alternatives, namely 7 plies of 36-ounce or 8 plies of 32-ounce duck, there would be a saving of about the same amount. Any kind of tandem drive would, however, require more machinery and more supports, the cost of which should be charged against the saving in the cost of the belt, but there is this difference to be taken into account—the machinery and supports are paid for only once, whereas the saving in the price of the belt is repeated every time the belt is renewed.

A Better Way to Do It. A reconsideration of the problem of carrying 330 tons of stone over a level distance of 1420 feet will show that the high tension in the belt as discussed above is due to its low speed, and the low speed was assumed because the 36-inch belt would carry the tonnage at that speed, and the 36-inch width was chosen to suit the large pieces of rock. If the rock could be reduced to 3- or 4-inch size the belt could be narrower, and hence the conveyor cheaper. A 24-inch belt at 350 feet per minute will carry nearly 350 tons of 4-inch

stone per hour. The horsepower for the empty conveyor would be less than that for a 36-inch conveyor, but the power to carry 330 tons of stone over a level distance of 1420 feet would be the same regardless of belt width.

To simplify the problem assume that the conveyor takes 30 horsepower whatever the width. Thirty horsepower means a net effective pull of 2830 pounds at 350 feet per minute, and, with a wrap of 210° on a single-drive pulley, the maximum pull in the belt becomes 3900 pounds. This is less than half the pull in the 36-inch belt at 150 feet per minute for the same work.

From Table 6-A it appears that a 24-inch 6-ply belt of 32- or 36-ounce duck would support the load properly; the unit stress would be 27 pounds per inch per ply, and from Table 6-B we see that a 32-ounce duck run over normal pulleys would not be stressed beyond the permissible limit, which is 27 pounds. This means (Table 6-C) that we could use a 42-inch head pulley, 24-inch snub pulley, and 36-inch for the other pulleys. It would not cost much more to make the head pulley 48-inch and get a belt wrap of 240°.

Design of Long Conveyors. In the example discussed above the belt and idlers for a 24-inch conveyor would cost about 30 per cent less than for a 36-inch conveyor. Other examples could be given; they would all show comparable results and would point to the following conclusions:

1. Make the belt no wider than capacity requires; if the presence of large pieces would require a wider belt, consider screening out the oversize and crushing it.
2. Since it is important to give the belt a high capacity rating, base calculations of capacity on 30° surcharge of load (see Table 8-B) or even larger if the belt can take it safely. Capacity ratings given in this book are conservative.
3. Run the belt fast. Use a feeder to load it evenly. If a number of belts are in series, consider making the one at the loading point wider and running it slower.
4. Use a single-pulley drive if possible. Make the head pulley larger than normal, and arrange the snub pulley to get the greatest possible wrap on the driver. If calculations show a belt stress too high for such a drive, use a two-pulley tandem drive.
5. If the conveyor is to be used under conditions which favor a long life for the belt, use the tensions recommended in Table 6-B, but if it is to be used for only a few years, it will save money and may be satisfactory to work the belt at a higher unit tension. On this point consult the belt makers; they are always making improvements in design and construction and the trend is toward stronger belts and higher working stresses.

6. Place a weighted take-up near the driver so that the weight will act promptly and with certainty to maintain the full amount of slack-side tension at the driver.

Ready-made Designs. The tables of dimensions of "standard" or "typical" conveyors shown in catalogs of some manufacturers serve a useful purpose in settling preliminary questions about cost, space required, etc., quickly and without calling on the services of a draftsman. But it is not safe to use the catalog diagrams for details of construction; they should be regarded as suggestions only. The supports must be designed for stiffness as well as strength and with regard for the nature and position of the driving machinery, for the particular circumstances, and with proper clearances for the moving parts, and for safety of employes. Although the information given in some catalogs permits the user to buy the machinery as so much merchandise, every belt conveyor should receive the benefit of some engineering advice before the work goes too far. Manufacturers are able to give such advice and are generally willing to furnish prints certified for construction.

How Not to Do It. A large rock crusher at a lime plant, fed by side-dump cars, delivered to a 36-inch belt inclined between 20° and 21°, 250-foot centers, and run at 350 feet per minute. There was no feeder over the crusher or between it and the belt. Some of the product was required to be in large pieces with least dimension about 8 inches; hence the crusher was set to produce that size. When the cars were dumped the smaller stuff rushed through the crusher and onto the belt. On account of the high speed, 350 feet per minute, and the steep angle, the pick-up was bad, the rock did not acquire belt speed promptly, and the sharp corners cut and tore the belt. Once started up the incline, pieces of rock would roll back and jump off the belt. To prevent that, continuous skirt boards were then added for the length of the incline, and finally, to prevent the tail end of a load on the belt from sliding or rolling back, sets of pawls or stops were placed over the belt every 20 or 30 feet. Each set consisted of eight 3-by-½-inch steel bars set edgewise, pivoted about 3 feet above the belt and mounted so that the lower ends of the bars could yield or move upward with the travel of rock up the incline but be rigid in the opposite direction to prevent any downward movement. Then a reciprocating plate feeder driven from the foot shaft was interposed between the crusher and the belt. When this installation was inspected it had been in service about six months, the cover of the belt showed signs of severe cutting and scraping, several sections had been cut out and replaced by new belt, and belt fasteners were used to hold the belt together where it had been split

lengthwise. The belt had been a good one, apparently 6-ply with a cover at least ⅛-inch thick.

The angle of incline in this case should have been 17° or less and the speed of the belt not over 200 feet per minute. This would have provided a capacity far in excess of the 1000 tons per day required and, with a feeder, would have allowed for some irregularity in the feed to the crusher without reducing the tonnage per hour.

CHAPTER 7

TENSION AND TAKE-UP DEVICES

Take-ups for Belt Conveyors. A take-up does two things: it removes the accumulation of slack in the belt, and it permits the belt to be stressed to the tension at which the pulley will drive it. If the take-up is of the weighted or automatic type (Fig. 7-1) it must carry enough weight to load the empty belt to the tension T_2 at which the pulley can pull the load; thus, if a belt driven by a wrap of 210° on a lagged pulley has a total tension T_1 of 2000 pounds on the pulling side, the idle tension T_2 is $\frac{2000}{3.61} = 550$ pounds (see page 145) and when arranged like Fig. 7-1 the suspended weight of the pulley and its attachments must be 1100 pounds to give the belt the necessary tension. By the same reasoning, if the weight is applied to a foot-take-up, it must be twice the amount of the idle tension T_2.

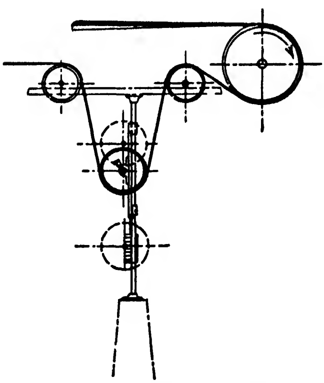


FIG. 7-1. "Gravity" Take-up for Belt Conveyor.

A suspended or "gravity" take-up can be placed anywhere on the return belt. It requires little attention and keeps an even tension in the belt. The objection to it is that it gives the belt three extra bends.

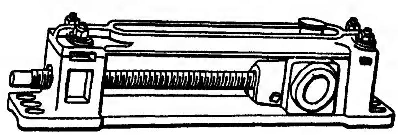


FIG. 7-2. Screw Take-up.

A screw take-up (Fig. 7-2) is simpler and cheaper and when placed at the foot of the conveyor it requires no extra bends in the belt. In the hands of a careless man, a screw take-up may pull a belt much tighter than is necessary for driving contact, and thereby injure it or pull the splice apart. On the other hand, if the accumulation of stretch is not removed as it forms, the slack tension at the driver may become too low; then, when the belt slips, it may be scraped or torn on the pulley side.

In a tandem pulley drive the ratio of $\frac{T_1}{T_2}$ is high, and for a given horsepower pull, T_2 is comparatively low, hence the belt coming from such a drive may be under very low tension. When such drives are near the foot of a conveyor it may be convenient to let the belt hang slack between the driver and the foot; then with screw take-ups at the

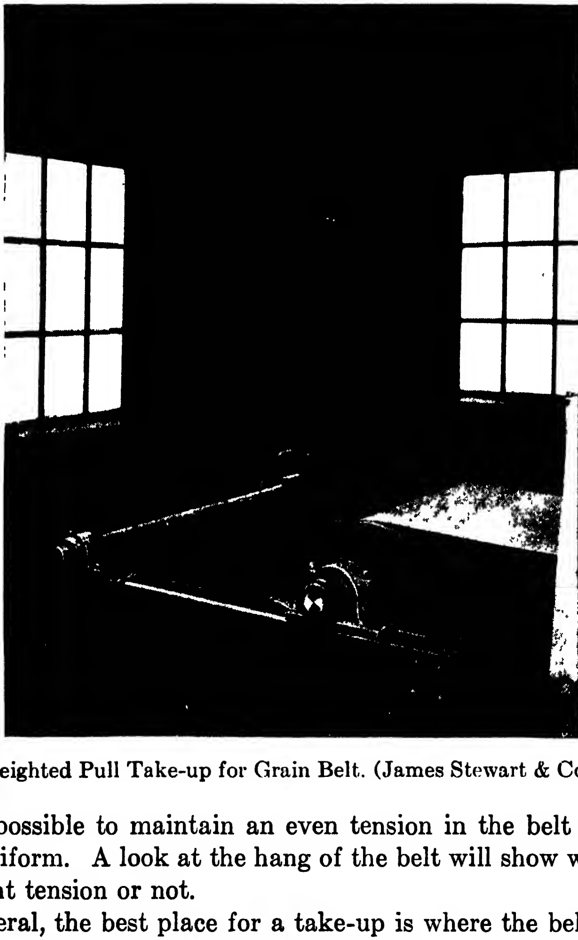


Fig. 7-3. Weighted Pull Take-up for Grain Belt. (James Stewart & Company, Inc.)

foot it is possible to maintain an even tension in the belt by keeping the sag uniform. A look at the hang of the belt will show whether it is at the right tension or not.

In general, the best place for a take-up is where the belt tension is low. One disadvantage of driving a conveyor at the foot pulley is the difficulty of disposing of the slack belt. (For an instance of this, see page 150.)

Weighted Pull Take-ups. Page 153 shows a take-up for a 60-inch 12-ply belt placed next to the driving end of a conveyor over 1000-foot centers.

Fig. 7-3 shows the foot of a grain conveyor with an automatic

take-up. The sheave in the corner is mounted on a pipe shaft to which are fastened two ropes leading to the shaft bearings. The weight rope makes a number of turns around the sheave, and as the weight descends, the take-up ropes are wound upon the pipe under a tension equal to five or six times the weight.

Take-ups for Long Conveyors. Suppose that a 500-foot conveyor, Fig. 7-4, has a take-up with weight enough to stress the belt at B and C to 10 pounds per inch per ply. In the upper diagram representing the conveyor at rest, various friction losses cause this unit tension to fall off toward the other end of the conveyor to, say, 7 pounds per inch

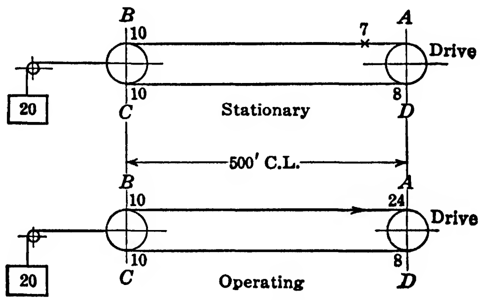


FIG. 7-4. Variation of Tension in a Belt Conveyor. Numbers in figure equal the tension in pounds per ply inch.

per ply at A and 8 pounds at D. In the lower diagram, when power is applied to the drive end, the 8 pounds slack tension at D enables the pulley to put 24 pounds unit tension into the belt at A, and the average unit tension in the 500-foot top run which was $\frac{10 + 7}{2} = 8.5$ pounds becomes $\frac{10 + 24}{2} = 17$ pounds. That is, by the act of setting the belt in motion the top run is stretched by an amount which corresponds to a unit tension of 8.5 pounds $(17 - 8.5)$ per inch per ply. From Table 3-D, page 57, the percentage of stretch for 8.5 pounds unit tension for, say, 28-ounce duck, is 0.45 per cent. The 500 feet of belt becomes $500 \times 0.0045 = 2.25$ feet longer, the take-up weight draws the pulley back one-half of this distance, or 1.125 feet, and the tensions at A, B, C, and D are not affected by the movement of the take-up. If, however, a screw take-up is used, stretch in the belt reduces the unit tensions, and if the take-up is not adjusted by hand, the slack-side tension, first at C and then at D may fall to a point where the pulley cannot drive the belt.

When power is shut off, a weighted take-up yields automatically

with the contraction of the belt as it changes from the operating to the stationary condition, but a screw take-up cannot do that. If it has been adjusted to suit the operating length of the belt and if it is not slacked off for the belt at rest, the shortening of the belt may cause harmful stresses and even damage to the belt and splice.

A gravity take-up is good on any belt conveyor; on long conveyors it is a necessity. See also page 179.

With a gravity-type take-up it is easy to introduce a definite amount of tension into the belt. With a screw type the tension introduced is increased or decreased by turning the screw to the right or left, but there is no indicator to tell the operator how many pounds' pull he is putting into the belt. When the path of a conveyor includes a vertical curve the definite amount of tension introduced by the gravity type is a decided advantage. The radius of the vertical curve is re-

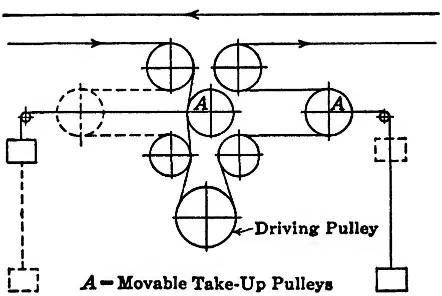


FIG. 7-5. Double Counterweighted Take-up for a Reversible Conveyor.

lated to the belt tension, a part of which is supplied by the take-up. If this latter tension is greater than the calculated pull the effect will be to lift the belt from the supporting idlers on the curve.

Another advantage of the gravity type on long conveyors is that the length of movement can be chosen to suit the conveyor. An increase in length of movement adds very little to the cost. The screw type is not usually made with a movement greater than 4 feet. Screw take-ups can be made with longer movement, but the parts, particularly the screw, increase in size and the cost mounts rapidly.

Gravity Take-ups on Reversible Conveyors. Fig. 7-5 shows a method of using two take-ups so that one of the counterweights will always put tension into the slack side of the belt. Another way is to mount the driving end of the conveyor on a wheeled carriage which is pulled back by a suspended weight; Fig. 7-6.

Length of Take-up Travel. If the belt is spliced with a metal fastener, 1 to 1½ per cent of the length of the conveyor is ordinarily

provided, with 12 inches as a minimum. If the belt has a vulcanized splice, the travel should be longer than 1½ per cent if there is room for it; not that the belt will stretch more than one with a metal fastener, but the cost of cutting the belt and making a new vulcanized splice is considerable. In one installation of long 48-inch belts, the ends of all the pieces were scarfed at the factory before shipment, and vulcanized joints were made in the field except that the last splice had

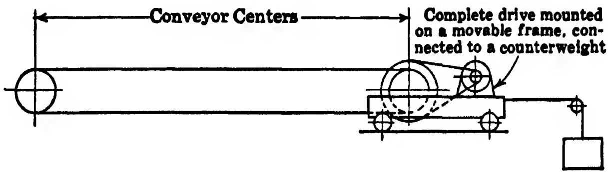


FIG. 7-6. Single Counterweighted Take-up for a Reversible Conveyor.

metal fasteners. It was intended that, when the belt had stretched to the limit of take-up travel, the metal splice would be discarded and replaced by a vulcanized splice. Actually, however, the belts did not stretch enough to use up the 6 feet of take-up travel provided, and when after several failures of the metal splice it was decided to substitute a vulcanized splice, it was found necessary to insert a piece of new belt long enough for a splice at each end.

CHAPTER 8

LOADING THE BELT

The carrying capacity of a troughed belt depends in the first place on how the material can be piled on the moving belt and safely carried. When the safe-load cross section has been determined by theory and confirmed by practice, the amount that the belt will carry is given by the formula below:

$$\text{Tons per hour} = \begin{array}{l}\text{Load cross section in square feet} \times \\ \text{Weight per cubic foot of material} \times \\ \text{Speed of belt in feet per minute} \times \\ \text{60 minutes} \div \text{2000 pounds per ton}\end{array}$$

The shape and size of the load cross section depends: first, on the way the material piles on the belt, that is, on the angle of slope of the material; second, on the clear margin that must be kept at the edges of the belt to prevent spill. It is not always possible to say in advance what angle of slope a certain material will take on a moving belt. The angle is always less than the natural angle of repose of the material when at rest, because the slight agitation of the load in passing over the troughing idlers causes the top of the pile to sink, and that flattens the angle of slope.

The clear margin or edge distance which should prevent spill must have some regard to the nature of the material, the depth of the load, and its angle of slope. Considering the great variety of materials carried on belts and the different conditions under which they are handled, it does not seem possible, in any formula for belt capacity, to make the margin variable for various operating conditions. If the margin is made reasonably large for wide belts and not too small for narrow belts, we have done all that is possible toward establishing a rational method of calculating load cross sections. From what has been observed on many belts in actual service it seems safe to call the margin or edge distance $0.05W$ plus 1 inch, where W is the width of the belt; in calculations of load cross section this gives results which correspond closely with belt capacities attained in belt-conveyor practice.

Fig. 8-1 shows the actual measured cross section of the load of bituminous coal on a 48-inch belt traveling at 500 feet per minute.

The angle of slope on each side is about 40°, and the top of the pile is defined approximately by the arc of a circle. This belt carried 116 pounds of coal per foot, a load much larger than is usual for a 48-inch belt. It was made possible by a combination of favorable conditions: first, the coal had been wet to keep down dust in the mine, and it was moist enough to stand at a rather steep angle and not spill over when

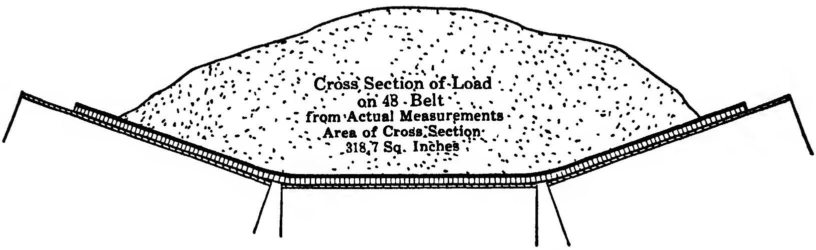


FIG. 8-1. Cross Section of Load on 48-inch Belt.

carried about 3 inches from the edge of the belt; second, the method of loading and the design of chutes and skirt boards received the most careful attention, and the operating conditions were under constant supervision in order to make the belt carry as much as possible.

Fig. 8-2, representing the outline of a belt on three-roll, 20° idlers, shows the method of calculating the load cross sections given in

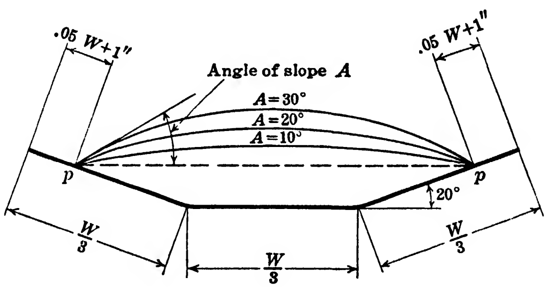


FIG. 8-2. Cross Section of Load for Various Angles on Which the Material Will Pile on the Belt.

Table 8-A. From points p at a distance of $0.05W$ plus 1 inch from the edge of the belt, lines are drawn at angles of 10°, 20°, and 30° with the horizontal to show the way the material may be expected to lie on the moving belt. Arcs drawn through p–p and tangent to the lines of slope mark the top of the pile. Each load cross section consists of a "level-load" portion the top of which is the straight line p–p, and a

"surcharge" portion defined by the arc corresponding to 10°, 20°, or 30° angle of slope. The areas are given in Table 8-A.

The arc marked 20° indicates the way many free-flowing materials will lie on the belt, or looked at in another way, it corresponds to a load cross section which is common with many materials under average operating conditions, that is, in everyday belt-conveyor practice. The arc marked 10° shows a load cross section for fine, dry, free-flowing materials which pile at a flat angle when they are on the

TABLE 8–A

Areas of Load Cross Sections

See Fig. 8–2.

1	2	3	4	5	6	7	8	9
Width of Belt, inches	Clear Margin, inches	Area of Level Load, square inches	Area of Surcharge			Total Area of Cross Section		
			10°	20°	30°	10°	20°	30°
			square inches			square feet		
14	1 7	7.56	3 06	6 20	9 50	0.074	0 096	0 117
16	1 8	10 4	4 19	8 47	13 0	0.101	0.131	0.162
18	1.9	13 8	5.49	11.2	17.0	0.134	0.173	0 214
20	2 0	17.6	6.95	14.1	21 6	0.170	0.220	0.272
24	2 2	26 7	10.4	21.1	32 4	0.257	0.332	0.410
30	2 5	43 7	17 0	34 3	52 6	0 421	0 542	0.669
36	2.8	64.9	25.1	50 7	77.7	0.622	0.803	0.991
42	3 1	90.4	34.7	70 3	108	0 869	1.12	1.37
48	3 4	120	46.3	93.6	143	1.16	1.48	1 83
54	3 7	154	58.8	119	182	1.45	1 90	2.33
60	4 0	192	73 5	148	227	1 83	2 36	2 91

belt. The arc marked 30° shows what can be done by careful attention to loading the belt and to the smooth and steady operation of the conveyor. Material like run-of-mine coal when the lumps are mixed with a mass of fine stuff will pile at an angle of more than 30° on a moving belt and not spill off if the edge distance is 0.05W plus 1 inch. Under the most favorable conditions, the carrying capacities may be still further increased as is shown in Fig. 8-1, but Table 8-A has not been extended to cover belts operated in that way. The load cross section under the 20° arc represents what can be carried under ordinary

conditions or what should be expected when the many contingencies and risks of everyday operation cannot be foreseen and allowed for in advance of actual operation of the belt conveyor.

Comparison with Old Formulas. A formula used in the early days of the business, about 1896, was:

Capacity of a troughed belt in cubic feet per hour at 100 feet per minute belt speed = $1.3W^2$ where W is the width of belt in inches. Another, in use about 1900, based on some experience, gave the capacity as $2W^2$, and some years later $3W^2$ and $3.2W^2$ were adopted. These increases in belt ratings came with the spread of knowledge of how belts should be loaded and operated, and also with a recognition of the fact that there is a gain in making a belt carry as much as it safely can, first, because a conveyor no wider than is necessary costs less for belt, idlers, and machinery; second, a belt that carries a full load cross section takes the wear and abrasion over nearly its whole width and costs less to replace than a wider belt that is worn out in only the middle of its width.

The formula, capacity equals $3.2W^2$, translated into terms of load cross section becomes:

$$\text{Load cross section in square inches} = 0.0768W^2$$

because $3.2W^2$ equals $\frac{0.0768W^2}{144} \times 100 \times 60$. This is not a rational formula because it makes the cross section depend only on the width of the belt and neglects the fact that the clear margin or edge distance not included in the load cross section need not be proportional to belt width. If 1.7 inches is a proper edge distance for a 14-inch belt, it is not necessary to have a margin of $1.7 \times \frac{60}{14}$ equal 7.2 inches for a 60-inch belt; experience has shown that a 4-inch margin is enough to prevent spill from a 60-inch belt if it is properly loaded and operated. This means that the term $3.2W^2$ underrates the capacity of wide belts.

This fact is recognized by the capacity formula in the "Handbook of Belting" of the Goodyear Tire and Rubber Company (first published in 1930), where the factor 3.2 in the old formula changes with the width of the belt. It is made to vary by steps from 3.2 for 12-inch to 4.0 for 60-inch belts according to $\frac{W + 180}{60}$ = factor. The Goodyear formula is a rational one and it has been widely used.

The old capacity formulas are deficient in another respect. They are based solely on the weight per cubic foot of the material carried, and say nothing about the angle at which it will pile on the belt or

the depth at which it can be conveyed. Foundry sand may not weigh much more than dry beach sand, but it will pile at a steeper slope and can be carried deeper on the belt. Ores will stand at a steep slope, but loose cement or dry sand forms a shallow pile. The capacities stated in Table 8-B are not hard and fast; they are based on what is known and has been observed, but in the choice of the arcs which form the surcharge in Fig. 8-2 they should be used with some discretion. The table suggests that, when conditions are right, belts can carry considerably more than the capacities given by the old formula $3.2W^2$. If the characteristics of the material are not fully known or if the operations are doubtful, then it is wise to "play safe" as to capacities. The old formulas took it for granted that most belts would not be regularly and fully loaded and might not be properly operated, and they gave values suited to those conditions. For example, in delivering coal to a boiler house, the capacity of a belt conveyor is measured by the weight of coal transported from the track hopper to a bin in the building. That depends not only on the load cross section carried on the belt but also on such items as delay in placing cars over the dump; delay in removing empty cars, trouble in getting coal, especially frozen coal, out of the cars, trouble with crushers, delay in changing position of trippers, screens clogging, chutes filling up, stopping for lubrication or machinery troubles. All these reduce the time of actual conveying; experienced engineers always make some allowance for them and choose a belt wide enough to give the required capacity in the net working time. For relation between boiler-house capacity, coal supply, and conveyor capacity, see Kent's "Mechanical Engineers' Handbook," pages *23*-46, etc., in the Wiley Engineering Handbook Series, Vol. 3.

Capacity of Belts Not Troughed. A flat belt will carry the material above line *p–p*, Fig. 8-2. These areas are given in Table 8-A, columns 4, 5, and 6. A flat belt will carry 45 per cent of the capacity of a troughed belt with an angle of slope *A* of 20° and about 55 per cent if the angle is 30°.

Capacity of Grain Conveyors. Flat belts for carrying grain are not used to the extent that they were formerly. Belts of modern manufacture are not hurt to any appreciable amount by the use of concentrator or troughing rolls when properly installed. For belts that take their load at one or two places only, it is common practice to space the carriers 5 or 6 feet apart, with troughing idlers alternating with plain cylindrical carrying idlers. But if a belt takes from a traveling hopper or from a series of chutes, spill or scatter of grain can be avoided by using all troughing idlers and spacing them not over 5 feet

TABLE 8–B

CARRYING CAPACITY OF TROUGHED BELTS WITH A UNIFORM FEED

See Fig. 8–2

Width of Belt, inches	Angle of Surcharge, degrees	Total Load Cross Section, square foot	Tons of Material per Hour (2000 lb.) at 100 f.p.m. for Material Weighing in Pounds per Cubic Foot					
			35	50	75	100	125	150
14	10	0.074	7.84	11 2	16.8	22 4	28.0	33.6
	20	0 096	10 0	14 3	21.5	28 7	35.8	43 0
	30	0.117	12.2	17.5	26.3	35.1	43.8	52.6
16	10	0.101	10 5	15.1	22.7	30.3	37.8	45.4
	20	0.131	13 7	19 6	29.4	39.3	49 0	58.8
	30	0.162	17.0	24 3	36 4	48.6	60 7	72.8
18	10	0.134	14.7	20.1	30.1	40 2	50 2	60.3
	20	0.173	18 1	25.9	38 9	51.9	64.8	77.8
	30	0.214	22 4	32 1	48 1	64.2	80 2	96.2
20	10	0.170	17.8	25 5	38 2	51 0	63.7	76.4
	20	0 220	23 1	33 0	49 5	66 0	82 5	99.0
	30	0 272	28.5	40 8	61 2	81.6	102	122
24	10	0 257	26.9	38.5	57 8	77.1	96 3	116
	20	0 332	34.8	49.8	74 7	99 6	125	149
	30	0.410	43 0	61.5	92.2	123	153	185
30	10	0.421	44 1	63 0	94 5	126	157	189
	20	0 542	57 0	81 5	122	163	203	244
	30	0 669	70 0	100	150	200	250	300
36	10	0.622	65.1	93.0	139	186	229	278
	20	0.803	84.0	120	180	241	300	360
	30	0.991	103	148	222	297	370	444
42	10	0.869	91	130	195	260	325	390
	20	1.12	117	168	252	336	420	504
	30	1.37	143	205	308	411	513	616
48	10	1.16	121	174	261	348	435	522
	20	1.48	155	222	333	444	555	666
	30	1.83	191	274	411	549	685	822
54	10	1.45	151	217	326	435	543	652
	20	1.90	199	285	427	570	712	854
	30	2.33	244	349	524	699	873	1048
60	10	1.83	191	274	412	549	686	824
	20	2.36	247	354	531	708	885	1062
	30	2.91	305	436	654	873	1090	1308

apart. The carrying capacity of grain belts is influenced to an important degree by the manner of loading: at the high speed at which grain belts travel, careless loading or irregular feed may cause a belt to carry much less than it should. When a grain belt is supported every 5 or 6 feet on troughing idlers of the style shown in Fig. 4-D,

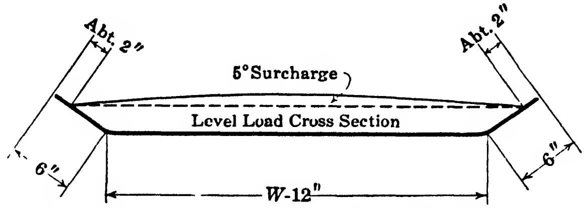


Fig. 8-3. Cross Section of Load on a Grain Belt Using 35-degree Concentrators.

about 6 inches width of belt is turned up at each edge. With a level load kept about 2 inches under the edges of the belt and a surcharge defined by an arc tangent to a 5° angle of slope, the load cross section has the shape shown in Fig. 8-3. The areas of the cross sections are given in Table 8-C, columns 2 and 3. The corresponding capacities in

TABLE 8-C

Areas of Load Cross Sections and Capacities of Grain Belts

See Fig. 8-3.

1	2	3	4	5
Belt Width, inches	Level Load Cross Section, square inches	Total Load Cross Section, square inches	U. S. Bushels per Hour @ 100 f.p.m. Bushels in Column 3	U. S. Bushels per Hour @ 100 f.p.m. in Average Practice
18	19.8	22.0	730	550
20	24.2	27.0	900	675
24	32.9	37.7	1250	1000
30	46.0	54.5	1800	1530
36	59.0	72.3	2400	2160
42	72.1	91.2	3000	3000
48	85.2	110	3700	3700

One cubic foot equals 1728 cubic inches
One U. S standard bushel equals 2150 42 cubic inches, or nearly 1.25 cubic feet
One British Imperial bushel equals 2219.36 cubic inches or about 1.03 U. S bushels
On any belt, capacity in U. S bushels per hour at 100 feet per minute equals average load cross section in square inches multiplied by $\frac{100}{3}$.

bushels per hour at 100 feet per minute are given in column 4. Belts working under good conditions of loading will give the capacities listed in column 4, but under ordinary or average conditions it is advisable to rate belts at the figures given in column 5.

Capacity of Inclined Conveyors. Within the safe operating angle and with cross sections not exceeding those shown in Fig. 8-2, inclined conveyors will carry practically the same tonnage as a horizontal conveyor. However, the potential capacity of an inclined conveyor is less than that of a horizontal conveyor. The material on an inclined belt tends to assume a flatter slope (angle A, Fig. 8-2), and that limits the cross-sectional area; if the conveyor is loaded on the incline, it is not so easy to pile a high load on the belt.

Peak Load Capacities. All rules for belt capacity apply to average conditions of loading where the feed is fairly uniform; but where a feeding device cannot be used or where the belt takes from a machine like a crusher whose rate of output for a period of some minutes may greatly exceed its average hourly rate, then the capacity of the belt should be considered on the *minute* basis and not the *hour* basis, unless the chute or hopper under the crusher is large enough to hold the excess of material during the time when the crusher output is above the hourly rate. (For an example based on this condition, see page 180.)

An excess of belt capacity over the average hourly rate should also be provided where the conveyor receives material dug from cars or boats by a grab bucket. When unloading begins from a full car or boat, digging is easy and the bucket will bring up more rapidly than later when the material gets low and the bucket cannot dig so well or be handled so fast. Under these circumstances the output at first may greatly exceed the hourly average.

There are also conditions which, in order to maintain a certain daily total, require conveyors to handle per minute or per hour loads much greater than average. These are frequently overlooked in discussing the capacity of the conveyor. The capacity should, of course, be equal to the "peak load" and not to the average rate.

Loading Chutes. Besides the normal duty of conveying material, a belt has to check the impact of material at the feed point and impart to it the belt's own velocity in the direction of travel. If 200 pounds of material falls from a height of 5 feet on a belt, the belt must absorb within its structure 1000 foot-pounds of energy in stopping its fall. It does it through the elasticity of its body of fabric and of its cover, if it has a cover. If there are hard sharp lumps, some of the energy of the falling mass is expended in cutting the belt or its cover, especially if the belt is so supported that it cannot exert its elasticity. This happens if

an idler is placed directly under the point of impact. To avoid it, the arrangement should be like Fig. 8-4. The idler *A* avoids the direct impact, yet it prevents the belt from sagging too far from the skirt boards *B*. If the belt were fed midway between two idlers, there might be too much deflection of the belt from impact, with consequent leakage sideways under the skirt boards. If the idler is directly under the point of impact, there is the added risk of breaking the idler.

If, in the case referred to above, the conveyor runs 5 feet per second, the work it does in bringing the 200 pounds from zero velocity to 300 feet per minute is $\frac{1}{2} \times \frac{200}{32.2} 5^2 = 78$ foot-pounds, and if the 200 pounds is fed on in 2 seconds, the equivalent is $\frac{78 \times 60}{2 \times 33{,}000} = 0.07$ horsepower for the capacity of 180 tons per hour. Though this represents only a small addition to the normal pull in the belt, its effect on the carrying surface can be understood if we imagine a grinding wheel made of the material carried, pressed against the belt and driven by 0.07 horsepower for every minute the conveyor runs. Of course, some of the material falls on itself and does not touch the belt, but the example will serve to emphasize the statement that the arrangement of the feed is a very important item in the design of the conveyor and that a poor feed may spoil a good belt.

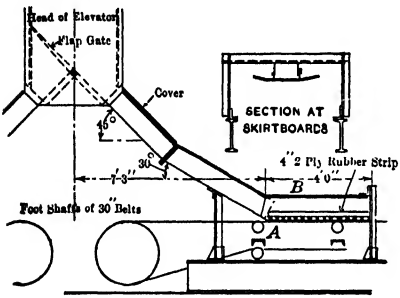


FIG. 8-4. Loading Chute and Skirt Boards.

Fig. 8-5 shows a bad arrangement of a loading chute; the rock hits the belt directly over the pulley; see page 122.

For a test to show the effect of impact at the loading point, see page 96.

Width of Chutes. The width of a belt depends, first, on the size of the pieces handled, and, second, on the capacity required. So far as capacity is concerned, the width can be determined from the speed and from Table 8-B, but the more important requirement is that the belt shall be wide enough for the largest pieces. This is really based on the necessity of having a loading chute that will not choke. A chute, to avoid spilling off the belt, can hardly be wider than $\frac{2}{3}W$ (W = width of belt), and since the width of a chute to avoid choking must be about three times the size of the pieces when they are uniform

in size, it follows that the belt width should be four or five times the size of the pieces where they are all of one size, as in sized anthracite coal and other screened materials. But run-of-mine coal, crushed rock, and such material handled on belt conveyors seldom consists of pieces of uniform size. For these, the width of the chute can be about twice the size of the largest pieces with a fair chance that no two of them will lock in position to block the flow.

FIG. 8-5. Bad Arrangement of Loading Chute.

Proper Angles for Loading Chutes. If belts could be fed with material moving at belt velocity, cutting and abrasion would be at a minimum; but it is seldom possible in practice. The best that can be done is to deliver material through a chute pointing in the direction of belt travel and at such an angle that the horizontal component of the velocity in the chute will be equal to belt speed. In practice, the best angle of chute can be determined only by trial; grain flows easily in chutes sloped 6 in 12 (27°) and rapidly at 8 in 12 (34°); but in Westmacott & Lyster's experiments in 1865 (see page 8) it appeared that the best angle for delivering wheat to a belt moving at 500 feet per

minute was 42½°. Anthracite coal in the domestic sizes flows readily in steel chutes at 5½ in 12 (25°), and in the steam sizes at 7 in 12 (30°), crushed soft coal requires 8½ in 12 (35°). Screened and sized stone or ore will flow in steel chutes at 7 in 12 (30°); the same material mixed with fine stuff requires 8½ in 12 (35°). For all these the slope of the loading chute should be 5° or 10° steeper, but for large lump coal or ore the angle should not exceed 45° (12 in 12); beyond that the lumps will not slide on the bottom of the chute, but descend in a series of jumps. Another factor with lumpy or sharp material is that in a steep chute the vertical component of its velocity becomes too great and

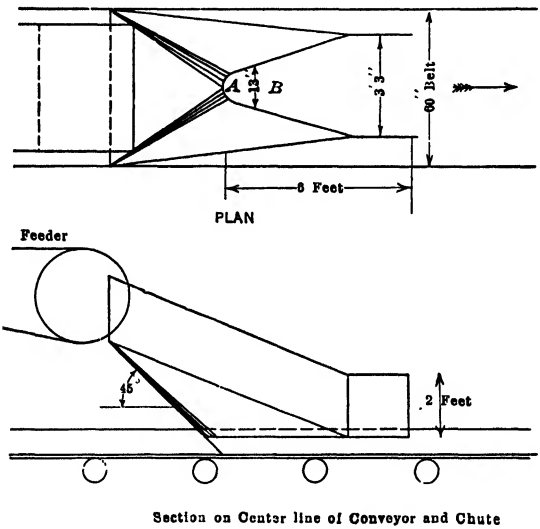


FIG. 8-6. Belt-loading Chute for Run-of-mine Coal.

the belt may be cut by the impact. To prevent that, the chute may be curved, or if the chute is long enough to give the material the necessary momentum, the angle may be broken, as in Fig. 8-4.

Both these methods are effective in preventing, in some degree, the effect of impact and in giving the material some velocity in the direction of belt travel.

A form of chute used in loading run-of-mine coal on a 60-inch belt is shown in Fig. 8-6. The rounded bottom does not interfere with the flow of fine coal or pieces of moderate size, but lumps of very large size are likely to be directed forward and upward by the converging corners so that they will strike the belt at B after it has already received a layer of the smaller coal at A.

Screen Chutes. From the earliest days of the business, efforts have been made to reduce the wear on the belt surface by screening out the fines in the loading chute and delivering them to the belt first, so as to cushion the fall of the lumps. Fig. 8-7 shows the idea, but it is not easy to avoid choking such a chute; even if bars are used instead of perforated plate, pieces stick between them and the chute chokes or else the velocity of flow through it is reduced. Another construction is to put finger bars on the end of the chute and let the fines fall between, while the lumps ride over, but this is open to the danger of pieces catching between the bars or under them and dragging on the belt.

When a belt is inclined at an angle close to 20° it is not easy to deliver material to it at belt speed; the lumps tumble around longer before becoming settled, and the wear on the belt surface is greater. In such cases a screen chute is a good thing if it is properly made. Figs. 8-8 and 8-9 show one designed to transfer run-of-mine coal from an apron feeder to a 36-inch belt inclined at 20°. The screening surface was about 30 by 50 inches, the bars were 4 inches by ½ inch spaced 3 inches; the spacers between the bars were set low to keep clear of lumps riding over the bars, and the apron plate *A* was set below the top surface of the bars for the same reason. Since the bars did not extend close to the belt, there was no danger that lumps caught between the bars would drag on the belt and injure it. The main discharge from the apron feeder fell on the screen bars while the dribble under the head wheel was caught in the side extension of the chute and put on the belt behind the screen bars.

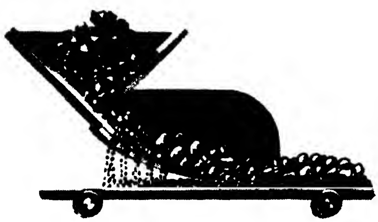


FIG. 8-7. Typical Screen Chute.

Fig. 8-10 shows a similar chute without the screen bars but with a curved bottom designed to catch the lumps falling over the head of an apron feeder and deliver them to a 36-inch belt at belt speed. Some of the discharge from the apron conveyor, including the drip, falls into the side extension chute and is put on to the belt before the lumps so that the large pieces fall on a cushion of small coal.

Transfer Chutes. In designing chutes to transfer from one conveyor to another, care should be taken that the discharge from the first belt cannot fall clear through onto the second belt without being guided by the chute, and also that the material is loaded *with* the run of the belt and not sideways. A belt loaded from the side will suffer from cutting and abrasion; it is likely to leak under the skirt boards,

and with an unsymmetrical cross section of load it will run crooked and need the assistance of side-guide or training idlers to keep it in place.

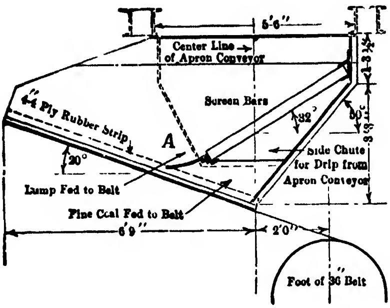


FIG. 8–8. Screen Chute for Loading an Inclined Belt Conveyor.

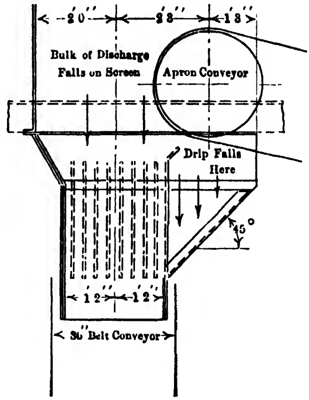


FIG. 8–9. End View of Screen Chute Shown in Fig. 8–8.

Loading on a Descent. Belts 36 inches and 40 inches wide, carrying run-of-mine coal down hill at angles of 15° and 16°, at a speed of 300 feet per minute, are successfully loaded on that angle of descent through screen chutes which deposit a layer of 1½-inch size and less before the lumps touch the belt. The skirt plates for these belts are about 16 feet long; after the coal leaves them the lumps are settled and do not roll.

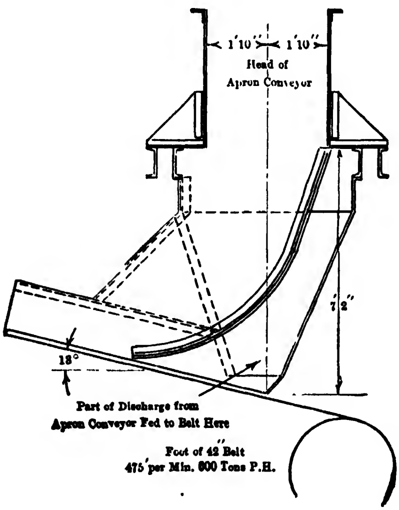


FIG. 8–10. Curved-bottom Chute for Loading an Inclined Belt Conveyor.

Horizontal Loading Ends for Inclined Belts. Conveyors inclined at about 20° for coke, large lump stone, and other difficult materials have been built with a curve at the foot of the slope, so that the belt is loaded on a short run which is nearly or quite horizontal. This arrangement permits the belt to be run at normal speed, there is less cutting and abrasion of the belt surface, and the material is not so likely to slip and roll while on the incline.

The curve should be so located with reference to the loading chute and be of sufficient radius that under no condition can the belt lift off the idlers and be cut by the skirt boards.

The danger of lifting the belt is avoided if the path of the belt is changed by the use of pulleys as in Fig. 1-15, but the scheme has several drawbacks: it takes up room; it gives the belt several extra bends, one of them with the dirty side of the belt against the pulley; and there is the added cost of three or four deflector pulleys with their shafts, bearings, and supports.

Skirt Boards. When a horizontal conveyor is loaded by a properly designed chute it should not be necessary to extend the sides of the chute more than 4 or 5 feet to confine the material while it is coming up to belt speed and to prevent lumps from rolling off. These extensions should be kept a few inches above the surface of the belt and the space closed by a strip of rubber belt to prevent fine stuff from scattering and lumps from getting under the edge of the skirt boards and jamming there. With steel skirt boards the risk of jamming is less than with wood. In loading run-of-mine or crushed coal onto belts 36 inches or wider, where the width between chute sides or skirt boards is not more than $\frac{2}{3}W$ (W = width of belt), the rubber strips can often be omitted, because then there is margin enough between the chute and the edge of the belt to prevent loss of material. Fig. 8-4 gives a design in steel construction. The chute shown in Fig. 8-6 has no skirt boards or rubber strips. For rubber strips, see page 206.

Damage from Skirt Boards. Nearly everyone in the belt-conveyor business can think of cases where belts have been split or seriously injured by skirt boards or chutes getting out of adjustment. To lessen the risk, skirt boards should be firmly supported. They should be no longer than is necessary, and there should be clearance enough under them to avoid cutting or scraping the belt if the belt, when running empty, should not lie down properly on the horizontal pulleys of the idlers. Means should also be used to prevent trippers from coming so close to the chute as to lift the belt under it (see page 235).

Skirt Boards for Inclined Conveyors. When the angle of inclination is greater than 10° the material does not acquire belt speed so readily, and lumps roll around more before becoming settled. For that reason, it is generally necessary to make skirt boards longer than for horizontal conveyors. If the incline is merely a short portion of the conveyor depressed to limit the travel of the tripper (see page 235) the boards are sometimes extended to the hump (Fig. 9-20) to prevent loss of material there when the belt flattens out in going over the flat-faced pulley. But unless the sides are made continuous with those at

the loading point, it is not safe to use skirt boards at such humps, because material is sure to catch under them. When belts are loaded to their standard ratings (Fig. 8-2) there should be no spill in passing over hump pulleys even though some of the material does describe a short trajectory and rise from the belt just as it passes the bend. In any case, it is better to reduce the load cross section than to use separate skirt boards at humps.

In order to prevent lumps from rolling off the belt, skirt boards have been run the full length of conveyors inclined at 20° or more, but it is not good practice. There is always the risk that the belt may move sideways and strike the skirt boards or their supports. If the slope is not too steep, skirt boards will not be necessary.

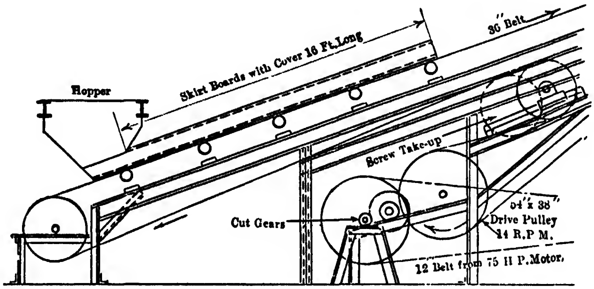


FIG. 8-11. Covered Skirt Boards for Dusty Material.

In feeding dry, dusty material to a belt it is sometimes advisable to extend the skirt boards and put a cover over them to make a box long enough to confine the dust. Fig. 8-11 shows such a guard for a 36-inch belt handling pulverized coal at a by-product coke plant.

Loading chutes are sometimes made with skirt plates to center the material on the belt. The distance between the lower edge of the skirt plates and the belt is covered by strips of rubber or belting; it is necessary that the skirt plates or strips should be cut to clear the troughed contour of the belt as determined by the idler used, and the chute and skirt plates should be supported in a manner that will definitely fix the position of the skirt plates with reference to the belt. The design of skirt plates and their supports is important and, if left to chance or to a millwright new to such work, the job is not likely to be done right.

Rubber Skirt Strips. Some rubber companies make skirt-plate strips of rubber without fabric insertion. These last longer than strips of rubber belting, and, since there is no fiber to pick up and hold abrasive particles, there is less wear on the conveying belt if the belt

should rub against the edges of the strips. Such material also makes good and durable lining for chutes.

Wear on Belts. The B. F. Goodrich Company's "Belt-Maintenance Manual," No. 2800, gives the following:

> The principal wear on a conveyor belt occurs at the loading points. As the material leaves the chute it should move in the same direction as the belt and as nearly as possible at the same speed. Loading at an angle with the travel of the belt increases the wear and may cause the belt to climb the idlers and damage the edge. The fall of lumps directly against the rubber surface causes not only wear, but also cuts, gouges, and breaks in the fabric. To lessen the wear, use a screen chute or a chute with a notched end; if the velocity of material is great, use baffle bars in the chute. Don't let material hit the belt directly over a pulley or an idler. If unavoidable, use idlers with rubber-covered rolls. Watch the clearance between belt and lower edges of chute and skirt boards. No stationary parts should rub against the belt. Keep the belt well supported at the loading point; don't let it chatter or vibrate, or else material will get between the belt and the stationary parts and hurt the belt. Heavy material requires idlers close together at loading point.
>
> In spite of careful placing, skirt boards often cause great wear on the belt cover. They should never touch the belt, empty or loaded. Strips of rubber at the bottom are useful to prevent scatter, but they should not bear hard on the belt. Design skirt boards so that clearances underneath increase in the direction the belt travels so that lumps will work free rather than jam tight.
>
> If a belt worth $3000 can be made to last 10 years instead of 2 years, the saving justifies considerable expense for efforts to avoid injury to the belt. . . . Rubber covers can stand up for a long time against abrasive wear and even impact of sharp material provided the rubber is not distorted beyond its elastic limit. . . . If heavy or sharp lumps drop hard on the belt, or if the belt scrapes against obstructions, or if pieces of material are caught between belt and pulley, then more harm can be done in a few days than in years of normal operation.

Feeding as Related to Capacity. The older formulas for belt capacity (see page 195) allowed for some irregularity in feed, and even for some bare places on the belt occasionally; but for the best results the feed should be under control so that the load cross section is uniform throughout the conveyor length. Table 8-A, "Carrying Capacity of Troughed Belts," assumes a continuous load cross section as shown in Fig. 8-2.

If the material is fine and lively a simple slide gate in a chute will control its flow, but this is not possible with lumps or with lumps mixed with fine stuff. For these some kind of feeder is necessary, especially

when the material is drawn from bins or from track hoppers. When crushers or pulverizers deliver to belt conveyors it is usually best to put the feeder over the crusher and then provide a hopper or a chute beneath it large enough to equalize any momentary rush of fine stuff through the crusher.

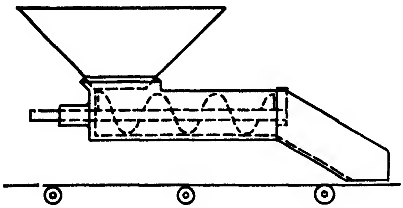


FIG. 8-12. Screw Feeder.

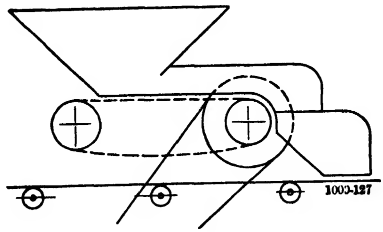


FIG. 8-13. Apron Feeder.

Feeders for Belt Conveyors. To pile an even load on the belt, the best type of feeder is one which has a steady forward delivery. The screw feeder, Fig. 8-12, will handle fine stuff, moist or dry. It can be

FIG. 8-14. Vibrating Feeder Loading a Belt Conveyor.

arranged to form an air lock if the material is delivered pneumatically to the bin above it. It is not long-lived in gritty material and is likely to be damaged if sticks, tools, or large lumps accidentally get into it.

The apron feeder, Fig. 8-13, carries the material, and is made in two types, steel apron or rubber belt. Steel aprons consisting of overlapping pans attached to chains are suitable for lump material. They can handle heavy material and the large lumps of coal, ore, and stone produced at the mine or quarry. Feeders for this class of work are rugged and heavy and consequently expensive. A disadvantage is that fine abrasive material, such as coke breeze, may get between the overlapping pans and cause wear.

The "apron" may be a conveyor belt (belt feeder) with the carrying run sliding on a steel plate or supported on flat or troughed idlers spaced about 12-inch centers. This type is well suited to handle fine material, even if it is abrasive and also lumpy material. Either apron type can be installed on an incline to save depth where desirable.

Vibrating feeders deliver a controlled uniform feed. Electromagnets furnish the rapid pulsations or vibrations which move the material along the trough. Fig. 8-14 shows one feeding dirt to one of the long belts at Grand Coulee, see page 272.

The Ross feeder (patented),* as used to deliver lumpy or mixed material to a belt conveyor, Fig. 8-15, consists of a chute in which a curtain of heavy endless steel chains, hung from a drum or tumbler, lies on the material in the chute and, by rotation of the tumbler and travel of the loops of chain, feeds the material from the chute. When the feeder is at rest, the curtain of chains is heavy enough to act as a gate to stop rushes of material; when the feeder is in motion, the material is fed to the belt under full control to the point of liberation by the chain apron. The chute can be made with a floor of screen bars so that fine stuff drops on the belt before the lumps hit it. The Ross feeder is made in a number of standard sizes to suit any width belt, will handle any material up to the largest size of quarried rock, and in quantities which can be fixed according to the revolutions per minute of the drum from which the chains are hung.

Some types of feeders deliver the material intermittently. The reciprocating feeder, Fig. 8-16, is simple and relatively inexpensive; it can be placed directly under a track hopper or a rock dump without danger of damage from falling lumps or from car couplings, mine props, tools, and such objects which sometimes come in run-of-mine coal. It does not give a uniform load on the belt because it seldom makes over 30 strokes per minute, but if it delivers to the belt through a long chute or through a crusher, its intermittent feed is made more uniform.

The shaking feeder, Fig. 8-17, is a modification of the reciprocating feeder; its strokes are shorter and more rapid, and the load cross sec-

* Ross Screen and Feeder Co., New York City.

tion on the belt is likely to be more uniform. These two feeders were the forerunners of the vibrating feeder, Fig. 8-14.

Rotary feeders, consisting of vanes turning in a vertical plane inside of a suitable housing, also circular plates revolving in a horizontal plane, are sometimes used.

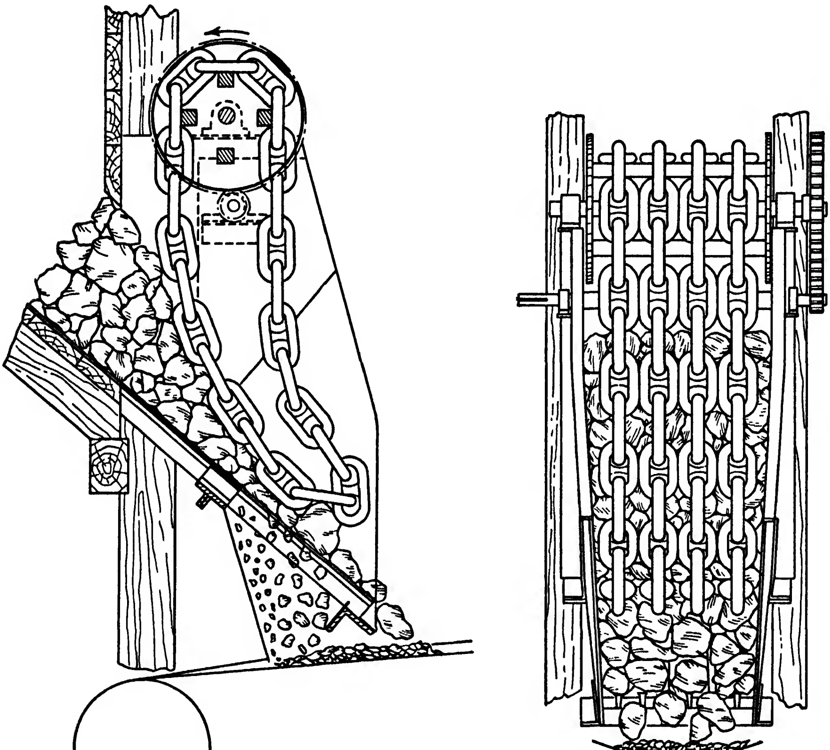


Fig. 8-15. Ross Feeder. (Ross Screen and Feeder Company.)

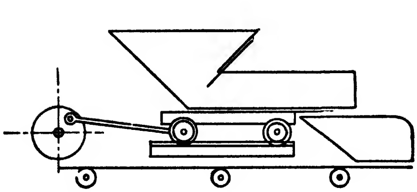


Fig. 8-16. Reciprocating Plate Feeder.

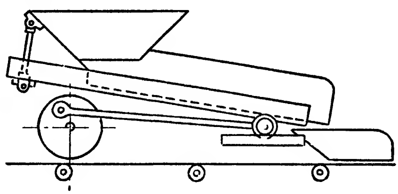


Fig. 8-17. Shaking Feeder.

No mechanical feeder will pile a perfectly uniform load from end to end of a fast-moving belt. The rate of feed per minute or even per second may be uniform, but the variation of feed within a second may

be enough to leave thin or bare places on a belt traveling 10, 8, or even 4 feet per second.

The Stuart patent 1,175,190 of 1916 covers the idea of using two or more belt feeders in series to deliver to a high-speed conveyor belt. The first feeder receives from a hopper or other source of supply at low speed; the second feeder receives from the first, runs at higher speed, and delivers to the conveyor belt.

Other Loading Devices. Several throwing devices have been suggested for delivering material to belt conveyors with a velocity that will lessen or avoid abrasion of the belt surface. In 1912 Thomas A. Edison patented a paddle-wheel feeder for use with fixed belt trippers where it was necessary to carry the material past the tripper. It was intended to remove the great objection to a series of fixed trippers, i.e., the wear on the belt from repeated reloading. The scheme was tried once on pulverized cement and discarded after a day's run. Similar methods are disclosed in the two Reinecke patents of 1911. Neither of them has come into practical use; they are costly and complicated and would take up more room than is necessary for a good chute. The cheapest and best way to speed material up to belt velocity is to let the force of gravity do it. There are, of course, places where chutes must be short, but it is well to remember that the expense of making a pit a foot deeper or an elevator a foot higher is often very much less than the continuing cost of the added wear on the belt.

Maxim. It may be set down as a maxim that the proper duty of a conveyor belt is to convey, not to speed up, material.

Traveling Loading Hoppers. When a belt draws its supply from more than one point, as from a number of openings under a long bin or from a number of bins, there is a choice between using a number of chutes or a single traveling hopper. In handling grain it is sometimes possible to use a number of fixed chutes, set with some clearance over the belt and with a pair of concentrators (Fig. 2-8) at each to prevent scatter and spill; but with coarser and more abrasive materials, fixed chutes with their skirt boards would interfere with the flow of material past them and would be likely to damage the belt. The chutes can sometimes be arranged to swing clear of the belt when not in use (Fig. 4-14), but it is frequently more satisfactory and often less expensive to use a traveling loading hopper.

Such hoppers for grain are comparatively light and can be pushed by hand. They consist of a box or funnel mounted on four wheels which travel on a track to which they can be clamped. The belt-loading chute is fixed at the proper distance above the belt and there are four concentrator pulleys carried on pivoted arms which by means

of a hand lever can be thrown into the operating position, or else thrown over to clear the fixed concentrators of the conveyor when the hopper is moved.

Traveling hoppers which draw coarse materials from bins are usually fitted with a feeder, because when the bin gate is opened wide it is often impossible to control the flow by hand so as to deliver a uniform load to the belt. In some cases the traveling frame is propelled by a motor which also drives a feeder of some kind. Fig. 8-18 shows a different kind where the conveyor belt furnishes the power for the traveling hopper. The belt passes around two pulleys at the end of the frame as in a tripper, and, from a shaft driven by paper friction wheels on one or the other of the pulley shafts, power is taken by chain drives to an axle under the frame and to the head of the apron conveyor.

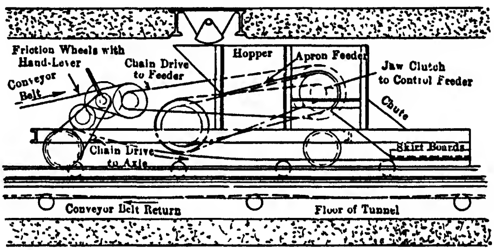


FIG. 8-18. Traveling Hopper with Feeder Taking Power from the Conveyor Belt.

A jaw clutch on the latter allows the feeder to be thrown out of gear during the traveling motion (Robb patent, 1904).

Angle of Inclined Conveyors. The slope of an inclined conveyor is usually limited by the tendency of the material to roll downhill; hence, screened or sized material cannot be carried on angles as steep as where the lumps in a mixture rest on a bed of fines. An intermittent feed is objectionable when the angle approaches the maximum; single lumps fed to the belt may not be picked up promptly, but may tumble around between the skirt boards for a time until a flat place happens to rest on the belt. The tail end of an intermittent feed may, for lack of a backing, become detached on the incline and dance up and down on the belt until the feed is resumed, or perhaps it may roll off. Conditions like these have fixed the angles at which it is practicable to convey various materials. Table 8-D is based on belt-conveyor practice.

In general, these angles are 10° or 15° less than the slopes at which the material will rest on the belt without moving. They may be determined by experiment with some of the material and a piece of belt,

TABLE 8-D

Permissible Inclines for Belt Conveyors

Material	Condition	Slope in Degrees
Briquettes. .	Egg-shape	10
	Cushion-shape	12
Cement. .	Dry	20–23
Clay.	Dry lump, loose	15–18
	Wet lump	18
	Dry fines	22
Coal, anthracite	Sized	16–17
bituminous . . .	Run-of-mine	18
	Slack	20–22
	Slack, moist	22
	Sized, small	17–18
	Sized, large	16–17
Coke	Run-of-oven	18
	Screened	17
	Breeze (fines)	20
Concrete	6-in. slump	12
	4-in. slump	20–22
	2-in. slump	24–26
Earth . . .	Loam, dry and loose	20
	Loam, moist	20–23
Feldspar . .	Dry and fine	18
Glass batch	. . . . . .	20–22
Grain	. . . . .	15–17
Gravel.	Run-of-bank	18–20
	Washed and screened pebbles	12–14
	Screened	15
Lime	Powdered	22–23
Ore, iron	Lump and fines mixed	18–20
	Sized lumps	18
	Fines	20–22
Salt	Coarse	18–22
Sand	Silica, dry	15
	Voids filled with water	15
	Run-of-bank	20–22
	Foundry, tempered	24
	Mixed with gravel	18–20
Slag.	Furnace, fines	20–22
Stone.	Crushed and sized	18
Sulphur	Lumps and fines	18
	Powdered	20–22
Wood	Fresh chips	27

but it is well to remember that, when the angle of incline approaches the maximum for any material, there is danger that the material may at times slip. This may be due to irregular loading leaving bare places on the belt, to difference in the amount of moisture in the material, or to the depth of load carried.

If there is a doubt about the angle at which a conveyor will carry material it is a wise precaution to make a test, particularly if the action of the material on a conveyor is not well known. The condition of the material, the shape and size of the lumps, the percentage of fines, and the amount of moisture are important. One of the most serious mistakes that can be made in the design of a belt conveyor is to make the angle too steep. See page 167.

The condition of the belt surface also has an effect on the way it picks up the load; if it is covered with fine dry dust, or if it is a new belt with the sulphur "bloom" still on it, or if on a cold day it is covered with minute frost crystals, the material may slip on it and refuse to go up the incline. Cleaning off the dust or getting rid of the frost by sprinkling salt on the belt are simple remedies which sometimes cure the trouble, but sometimes the angle is so steep that there is nothing to do but roughen the belt surface by fastening strips of belt on it or driving clinch rivets through it. As a general rule—"better be safe than sorry"—keep the angle of incline less than the maximum.

Special Belts for Inclined Conveyors. A number of United States patents have been granted on belts with rough or corrugated surfaces, intended to prevent slip of material and permit the conveyor to be run at an angle steeper than normal. The principal reason for making a conveyor steep is that it requires less floor space. A conveyor with a lift of 20 feet on a 20° slope takes 55 feet of floor space and is 60 feet centers but requires only 35 feet of floor space and 40 feet centers if the angle is 30°. Some of these special belts are open to the following objections: (1) They bump over the return idlers. (2) They cost more than ordinary belts. (3) When they carry fine stuff, particles lodge in the roughness or the corrugations do not discharge clean, cannot be removed by brush or scraper, and hence drop off on the return run. In spite of these drawbacks, there are places where a rough-top belt is useful. For their use in package handling see page 261.

In handling very gritty substances it is better to keep the angle well under the maximum at which it is possible to carry them without roll or visible slip, because in passing over the idlers there is an invisible slip or rearrangement of load which at steep angles scours the face of the belt. Short steep belts handling gritty ore have been

known to fail very rapidly from destruction of the rubber cover due to this invisible slip of material.

Choosing a Safe Angle of Incline. In preliminary layouts of inclined belts it is well to assume an angle safely under the maximum; then there will be some leeway to increase the angle should it become necessary during the development of the details. Those experienced in such matters know that in the final drawings the foot of an inclined conveyor is generally lower and the head often higher than in the original plans. Both these changes increase the angle of incline unless the horizontal distance between terminals can be increased. It is often inconvenient or perhaps impossible to do that, since at slopes around 18° or 20° a foot of vertical height corresponds to 3 feet of horizontal distance, and to change the height 1 foot without altering the angle means that the terminals must be separated horizontally 3 feet or more.

Much of the trouble with inclined belts comes, not from choosing the wrong angle for the material carried, but from the fact that in the development of the plans the angle becomes steeper than was at first intended.

Change in Practice. In the early days of belt conveying some engineers used angles of slope up to 6 in 12 (26½°) in order to have short conveyors or avoid elevators. Sometimes the conditions of feed and of the material, crushed coal, crushed rock, etc., were such that the conveyor worked satisfactorily, but more often there were difficulties and disappointments. In recent times practice has changed and angles greater than 20° are now rare.

Speeds of Belt Conveyors. The great advantage of belt conveyors over other forms of continuous conveyor is that they can run at higher speeds without increase of noise or much greater friction losses. So far as the transfer of material is concerned, all belt conveyors could, like grain conveyors, be run up to the speed at which the material would be blown off the belt by the resistance of the air. In practice, however, other factors govern the speed. The width of the belt and its speed are, of course, determined primarily by the capacity required; but, in addition, the loading conditions and the size and characteristics of the material must be considered.

If the material is not lumpy or abrasive, especially when it is free flowing like grain or crushed coal and the supply is regular as from a large bin, the belt may be run fast with the maximum speeds shown in Table 8-E. Under these conditions a loading chute can be arranged to deliver a uniform load to the belt. As the lumps increase in size it becomes increasingly difficult to maintain a uniform feed. A feeder of

the apron type, Fig. 8-13, will deliver uniform quantities of material per minute; but the fall of lumpy or mixed material over the head of the feeder may vary from second to second, and with a belt moving 8 to 10 feet per second it is impossible to avoid a thin load or even bare spots at intervals on the belt. When handling lumps approaching the maximum size on belts 30 inches and over, the speeds in Table 8-E should be reduced about 15 per cent.

TABLE 8-E

MAXIMUM ADVISABLE SPEED AND SIZE OF LUMPS

Width of Belt, inches	Maximum Lumps, inches		Maximum Speed, feet per minute	
	Sized	Unsized	Non-abrasive Fine Materials	Grain
14	2	3	300	400
16	$2\frac{1}{4}$	4	300	450
18	3	5	400	450
20	$3\frac{1}{2}$	6	400	500
24	$4\frac{1}{2}$	8	500	600
30	6	11	550	700
36	8	15	600	800
42	10	18	600	800
48	12	21	600	800
54	14	24	600	800
60	16	28	600	800

NOTE: "Unsized" means a mixture of fines and lumps in which 90 per cent of the pieces are under the maximum size and 75 per cent are less than one-half that size.

If a material is sold or used as a sized product or is fragile, such as anthracite coal or sized coke, it is important to avoid breakage at the discharge point. The speed of the belt must not be so high as to discharge such material with violence at trippers or at end chutes. For illustrations of end discharge see Figs. 9-1, 9-2, 9-3, and 9-4.

Abrasive materials, particularly the lumps, even under relatively good loading conditions wear the cover of the belt; reducing the belt speed decreases the frequency of this wear and also reduces the wear on the discharge chutes. The maximum belt speeds given in Table 8-E should be reduced about 15 per cent for handling fine abrasive material and about 30 per cent for lumpy abrasive material.

Speeds for Grain Belts. The experiments by Westmacott and Lyster in 1865, see page 8, showed that oats, bran, and flour could be carried at speeds up to 480 feet per minute without being blown

off the belt by the air resistance; they found also that wheat could be carried at 540 feet per minute safely, but that the chaff in it would be blown off. These trials were made with belts 12 to 18 inches wide, run flat except for concentrators at the loading points.

In modern practice wheat and corn are carried at speeds up to 800 feet per minute on belts 36 inches and wider; but on belts less than 20 inches wide the speed should be less than 500 feet per minute. The main reason is that in chutes suited to wide belts the flow of grain is rapid and regular and the belts can be loaded full, but in narrow chutes the flow is not so fast and even, and the narrow belt running faster than 500 feet per minute will not take a full load nor will the loading be uniform along the run of the belt.

High Speed Not Practicable for Narrow Belts. Narrow belts should be run slower than wide belts because the material is loaded closer to the edges of the narrow belt and as the speed increases the material is more likely to spill over the edge. Narrow belts also have a relatively small contact with the horizontal roll of the troughing idler and are more difficult to "steer," i.e., to run straight. Increasing the belt speed does not help this condition.

Speed Limited by Idler Construction. Old-style idlers with cast-iron pulleys mounted on shafts and grease lubricated revolved about 60 to 75 r.p.m. for each 100 feet per minute of belt speed. The pulleys were not centered and balanced as carefully as transmission pulleys, and at belt speeds above 400 to 500 feet per minute were likely to be noisy and the idler stands would shake loose from vibration. With the general acceptance of the anti-friction-type idler improved manufacturing methods came into use. Modern idler rolls with anti-friction bearings run quietly and smoothly at high speed. At a belt speed of 600 feet per minute a modern 6-inch-diameter roll turns about 380 r.p.m. without signs of distress or too much noise. The improvement in idler construction has eliminated this former limitation of conveyor belt speed.

Possible Maximum Speeds. Two limitations on the speed of a belt conveyor are: the ability to load it, and the air resistance blowing the material off the belt. Conveyor speeds have been increasing; 48-inch belts handling run-of-mine coal may be run at 600 feet per minute and 30- to 42-inch belts are handling grain at 800 feet per minute.

Minimum Speeds. The discharge over a 24-inch-diameter pulley with a belt speed of 140 feet per minute is shown in Fig. 9-1. Even slower speeds will discharge clean into a bin or to a chute which starts on the vertical center line of the pulley and below the return belt.

It has been stated that, "under 150 feet per minute speed, the cost of the belt conveyor per ton of bulk materials handled, even with a minimum ply of belt, commences to be uneconomical as compared with other types of conveyors equally suited to the operating conditions." Although this expresses a correct relation between belt conveyors and other types of conveyors as to first cost and operating cost, there may be reasons more important than costs which make a belt conveyor at 100 or 150 feet per minute preferable to any other conveyor at any speed.

For speeds of belts with trippers see page 232.

Full Load Cross Section Desirable. The aim should be to run the belt no faster than is necessary to give the capacity under the conditions of loading. A full cross section of load at a slow speed means a deeper pile and proportionately less material in contact with the belt, hence less cutting and abrasion, less strain from pick-up of material, and less energy wasted in revolving shafting and gearing, foot shafts, snub shafts, tripper mechanism, etc. Fig. 8-19 shows a belt carrying only one-third of its normal rated load, and superimposed upon the light load, the other two-thirds of its normal load. Only a portion of the added load touches the belt. Actually the load is tripled with an increase of 60 per cent in the amount of belt surface in contact with the material. The full load distributes the load over more of the belt, the light load concentrates it in the middle; and when a belt is worn out in the middle, it is all worn out.

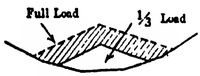


Fig. 8-19. Comparison of Full Load Capacity and One-third Capacity.

The rated capacities of belts are, as may be seen from Figs. 8-2 and 8-3, considerably less than what can be piled on the belt; where the feeding arrangements are favorable, it may be possible to get a larger load cross section than standard and hence a slower speed for the required capacity. After a belt conveyor has been in regular service for a short time it is always advisable to study the feed and the load, and then reduce the belt speed when it can safely be done.

Belt Speed as Related to Discharge. See Chapter 9.

CHAPTER 9

DISCHARGING FROM THE BELT

Discharge over head pulley is the simplest form of belt-conveyor discharge. The path of the material after it leaves the belt is a parabola; its appearance at various speeds is shown in Figs. 9-1, 9-2, 9-3, and 9-4.

To determine where the material leaves the belt assume that the belt is tangent to the pulley at a, Fig. 9-5. At that point material of weight W in pounds is acted upon by centrifugal force directed radially outward with a force $\frac{Wv^2}{gR}$ in pounds, where v is belt velocity in feet per second, g = acceleration of gravity = 32.2 feet per second per second, and R is radius in feet from the center of rotation to the rim of the pulley. At the same time the force of gravity acts vertically downward with a force of W pounds. The component of this force in a radial direction acts against centrifugal force. This component is W cosine A in Fig. 9-5. When it is greater than centrifugal force at a or b, the material stays on the belt; when it is less, the material is lifted from the belt with a force $\frac{Wv^2}{gR} - W$ cosine of angle and moves off in a trajectory.

Examples. 1. Suppose a belt traveling 5 feet per second meets a 24-inch pulley at point a 20° from the vertical. Centrifugal force $= \frac{W25}{32.2 \times 1} = 0.77W$. The opposing force due to weight is W cosine 20°, about $0.93W$. Since this is greater than centrifugal force, which equals $0.77W$, the material will not leave the belt.

2. If the belt just mentioned ran 6 feet per second, centrifugal force $= 1.11W$, which is greater than $0.94W$ and the material would tend to leave the belt at a.

3. If the belt speed is 7 feet per second and the pulley diameter 48 inches, centrifugal force $= \frac{W}{32.2} \times \frac{49}{2}$, which is less than $0.94W$ and the material will not lift at a.

In these three examples R and v have been taken at the rim of the pulley, which is not strictly correct. In example 1, if we say that be-

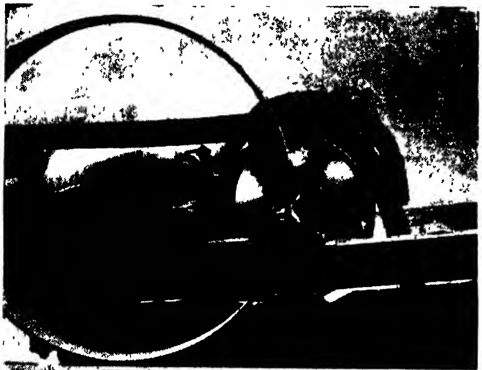


Fig. 9–1. Belt Speed 140 Feet per Minute, 24-inch Pulley.

Fig. 9–2. Belt Speed 275 Feet per Minute, 24-inch Pulley.

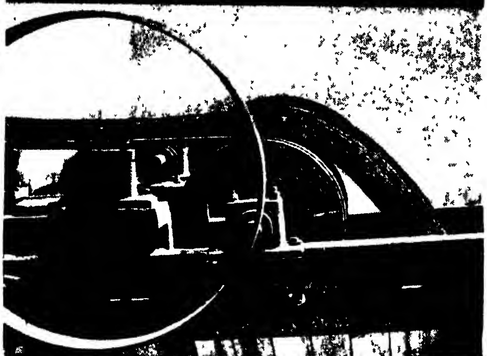


Fig. 9–3. Belt Speed 360 Feet per Minute, 24-inch Pulley.

Fig. 9–4. Belt Speed 780 Feet per Minute; 24-inch Pulley.

cause of the depth of the pile on the belt $R = 1.2$ foot and the corresponding velocity $v = 6.0$ feet per second, then centrifugal force $= 0.93W$. Since this is nearly equal to $0.94W$, the material is about ready to leave the belt but the action is not decisive. In a belt conveyor the effect of this is sometimes seen in the slight agitation of the load as it approaches a head pulley or a hump pulley.

If the material has not left the belt before it comes to point b it is there acted on by centrifugal force $\frac{Wv^2}{Rg}$ and a vertical force W, because at b the angle is zero and cosine $0° = 1$. If the two forces are equal they are in equilibrium and material is about to leave the belt; but if W is the greater, material will stay on the belt as in Fig. 9-1 until it comes to a point c, Fig. 9-5, where $\frac{Wv^2}{gR} = W \text{ cosine } B$. It then moves off in a trajectory.

To compare the action shown in Fig. 9-1 with theory, what is the angle at which discharge begins? Here $v = 2.33$ feet per second; the cosine of the angle is $\frac{2.33 \times 2.33}{32.2 \times 1}$; this equals about 0.17, and the angle B is close to 80°. What is the angle in Fig. 9-2? Here $v = 4.6$ feet per second; the cosine of the angle is $\frac{4.6 \times 4.6}{32.2 \times 1}$; this equals about 0.66, and the angle B is close to 49°. From the figures it appears that the discharges begin about where theory indicates. In Fig. 9-3, $\frac{Wv^2}{32.2 \times 1} = 1.11W$; and in Fig. 9-4, it equals $5.2W$. Both values are greater than W, and discharge begins at the top of the pulley; but in 9-4 it is more vigorous, and, since the velocity is higher, the trajectory is flatter.

The Trajectory. Its path is determined by (1) the place where the material leaves the belt—this is discussed in the preceding paragraphs; (2) the direction in which it moves at the instant it leaves the belt. As shown in Fig. 9-5, detail at a, this direction is the resultant of two forces. One force, tangential to the pulley, is that which gives the material a velocity v in the line of belt travel; it equals $\frac{Wv}{g}$. The other force radial in direction, acts only if centrifugal force due to high belt velocity or small pulley diameter is greater than W for a horizontal conveyor or greater than W cosine A for an inclined conveyor. Under such circumstances the material is lifted from the belt, starts off in the direction of the diagonal of the rectangle of forces, and ends in a curve like ax. But usually centrifugal force is not so great in

comparison with W; the material does not lift; it moves off tangent to the pulley and makes a trajectory like ay or az similar to those shown in Figs. 9-1, 9-2, 9-3, and 9-4.

To show the trajectory on a drawing proceed in this way: Draw the pulley and the approaching belt; determine the place where discharge begins, and from that point draw the line of direction in which discharge starts. This line extended shows the path which material would take if gravity did not act on it; mark on it points to show dis-

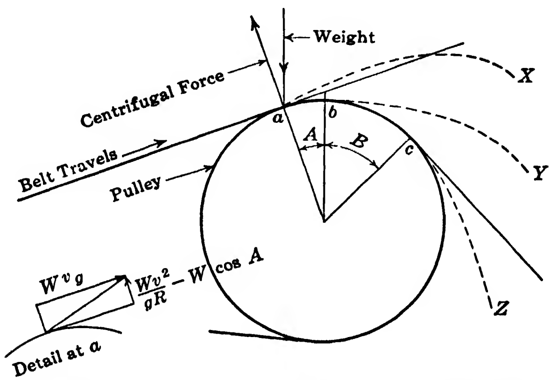


FIG. 9-5. Forces Acting on Material at Discharge.

tances traveled by the material at belt speed in short intervals of time for, say, each $\frac{1}{20}$ second. From these points draw verticals to mark the distance a particle of material would fall in $\frac{1}{20}$ second, $\frac{2}{20}$ second, and so on. If belt speed is 100 feet per minute, the distance moved in $\frac{1}{20}$ second is $\frac{100 \times 12}{60 \times 20} = 1$ inch, and that would, to the scale of the drawing, be the spacing between the verticals for each 100 feet of belt travel in feet per minute. The formula for the free fall of material, $D = \frac{gt^2}{z}$, gives the lengths of the verticals. For the first $\frac{1}{20}$ second it is $\frac{32.2 \times 12 \times 1}{20 \times 20 \times 2}$, about $\frac{1}{2}$ inch; for the second interval, 4 times as much, about 2 inches; for the tenth interval, 100 times as much, about 50 inches. These figures disregard air resistance and are only approximate, but they are good enough for practical purposes. They are the basis of the tables and diagrams published in trade catalogs.* The method of laying out the trajectory is shown in Fig. 9-6.

* E. P. O. Booth in *Engineering and Mining Journal,* Vol. 135, No. 12, December, 1934, discusses the discharge of dry crushed rock and its trajectory.

Position of Chute. In handling material easily broken, like sized anthracite coal, briquettes, or screened coke, it is desirable to place the chute so that it receives the discharge without shock and without excessive drop. At the same time, the top edge of the bottom plate of the chute should be set so that no pieces can jam in the gap between it and the belt, and in such a position that, if the conveyor should be stopped while loaded, the dribble of material over the pulley in slowing down or starting up will not fall into the gap and be wasted or stick there and damage the belt. To a great extent these requirements are contra-

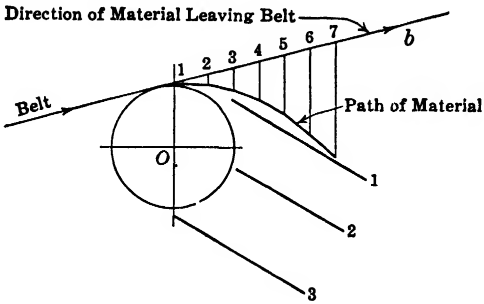


FIG. 9-6. Trajectory of Discharge from Belt Conveyor.

dictory; for an easy delivery the chute should be at 1 (Fig. 9-6), but for safety to the belt it should not be placed higher than 2; if the material tends to stick to the belt and is not hurt by the drop, it is better to put the chute at 3. If there is a cleaning brush at the discharge pulley the chute must be farther back, as in Figs. 10-4 and 10-5, so that at no time can material discharged over the head pulley fall directly on the brush.

Tripper Chutes. In trippers where it is important to save height, the chute is usually placed at position 2, and the brush is placed below it so as to throw the fine stuff clinging to the belt back on the empty run as it leaves the tripper.

Choked Chutes. When there is danger of material backing up in a choked discharge chute and then scraping the belt, it is well to set the upper end of the chute some distance below the pulley, so that when the material does back up, it will overflow from the top of the chute before it reaches the belt on the pulley.

Dribble over End Pulley. When a belt conveyor delivers to a bin through a tripper it is never safe to assume that all the material will be discharged at the tripper and none over the end pulley of the conveyor. The fine stuff which clings to the belt or which is brushed off in the

tripper falls over the end pulley, and if the latter is not placed over the bin, the dribble outside of the bin may be a nuisance. If it is necessary to place the end pulley beyond the bin, an auxiliary chute may be provided to catch the dribble.

Plows or Scrapers. When belts handle material like wood chips, fine coal or dry chemicals, which are not abrasive and which can be carried with little or no troughing, it is possible to discharge them at various places along the run by plows or scrapers set diagonally across the belt (Fig. 9-7). The lower edge of the plow should be a few inches above the belt, and the actual scraping of material should be done by a strip of belting fastened to the board and touching the conveyor belt. Scrapers are sometimes hinged at one end and bear against a stop at the other, so that they may be worked by a pair of pull ropes from a distant point; or they may be fitted between a pair of vertical guides as in the figure and transferred from place to place to change the discharge. With any construction, a chute or guard should be provided at each discharge point to prevent scattered material from collecting under the upper run and fouling the idlers or else falling on the return belt.

FIG. 9-7. Discharge from Flat Belt by a Diagonal Scraper.

Plows can be used on belts troughed with idler rolls inclined 20°. A flat steel plate under the belt at the plow flattens the troughed belt and the material can be scraped off. This type is frequently used to discharge molding sand in foundries. The steel plate and the plow can be mounted on rollers to give a movable discharge. Plows are also used in places where it is inconvenient or impossible to install a tripper.

In some European boiler houses the bunker is served by a flat belt which runs through a movable carriage equipped with a V-point plow and a two-way chute. The movement of the carriage is controlled from the floor of the boiler room by a hand-power winch operating a pair of pull ropes. An indicator on the winch shows the position of the carriage, and it is not necessary for a man to go aloft to change the position of discharge. Besides that, the carriage is cheaper than a tripper and takes up less headroom.

Objections to Scrapers. The general objection is, of course, the wear on the surface of the belt. A specific fault of the construction shown in Fig. 9-7 is that if the material is heavy, or the belt troughed, there is a component of pressure at the scraper which tends to force the belt out of line and off the idlers at the point of discharge. This can be avoided by making the scraper or plow symmetrical with a V point to discharge on both sides of the belt. With this construction, the discharge is not spread over so much length, but a disadvantage is that the ends of the rubber scraping strips which bear on the belt may catch at the belt splice or at worn places in the belt and do damage to the belt or else be worn off or torn loose.

Comparison with Trippers. As compared with a tripper, a plow causes more wear on the surface of the belt, but it does not tend to separate the plies of the belt by breaking down the friction layers between them by reverse bending, as a tripper with its small pulleys often does. If the material is fine and not abrasive, the life of the belt would ordinarily be determined by the life of its friction and not by wear on the surface. In such a case it may be ultimately more economical to use a scraper and put some wear on the belt surface so as to equalize in some degree the external and internal wear of the structure of the belt. Even though the life of the belt is shortened, it may still be economy to pay for belting rather than for a tripper and the necessary attendance, maintenance, and repairs.

Distribution to Separate Bins. When a belt is used to distribute material to a series of small bins a traveling tripper is at a disadvantage because too much time is lost in transferring the tripper from place to place and in making sure that the belt is empty when the tripper moves across the clear space between the bins, so that the tripper will not discharge material between the bins. There is also the expense of an attendant to set the tripper and make sure that it is clamped to the rails so tight that it will not work loose and travel with the belt. Fixed trippers are sometimes set over each bin, but if the service is hard and continuous they hurt the belt by the repeated reverse bending and by the repeated delivery of material back on the belt when the discharge point is beyond one or more of the trippers. The by-pass chutes furnished with trippers, traveling or fixed, do not deliver material back to the belt at belt speed, and the wear on the belt is greater than from a well-designed loading chute.

When fine material is to be distributed to a series of small bins, as in delivering foundry sand to the hoppers of molding machines, a number of scrapers operated by pull ropes will sometimes do the work at

less cost and with greater convenience than any arrangement of trippers, traveling or fixed.

Other Discharge Devices. An old device, patented by Palmer in 1888, is shown in Fig. 9-8. When the idler pulley is tilted, the material slides off the belt, but the discharge cannot be located with accuracy because it begins and ends as a dribble over the edge of the belt and the bulk of it is spread over some distance of travel. The scheme is used in a few places with belts less than 24 inches wide to discharge fine dry material into large bins where the position of discharge is not important. The return belt must be guarded against the spill from the upper run.

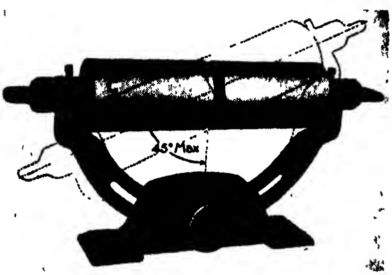


FIG. 9-8. Tilting Idler to Discharge Material from a Belt. (Jeffrey.)

A device similar to a plow is shown in the Brotz patent of 1912. It consists of a horizontal disc of steel plate mounted on a vertical shaft and extending over the belt in such a way that as the disc revolves it will take material from the belt and deliver it to one side. It was tried once in a foundry to remove sand from a flat belt to a series of small bins. The discs were driven by frictional contact with the belt and could be moved into or out of operating position. They were not satisfactory, and were finally replaced by scrapers set diagonally across the belt and controlled by pull cords from the floor below.

Discharge by Inverting the Belt. Discharge at points along the run of a belt conveyor can be effected by fixed trippers. If they are set in series to discharge at several points it is necessary to provide all but the last with a by-pass chute to reload material on the belt and fit them with flap gates to divert the stream, or else, as in Fig. 9-9, let the main chute fill up and overflow into the by-pass chute. This was the arrangement adopted in an American cement plant built about 1902 where a stockhouse was filled by a belt running through a long series of fixed trippers. The travel of the cement in the by-pass chute was so short that it did not acquire any speed, and hence it wore the belt. In applying, in 1907, for a patent on a rotating-drum device to replace the by-pass chutes on these trippers and to throw the cement back on the belt with some velocity in the direction of travel, the engineer of the plant says, "I find that the belts become very rapidly worn, since at

each discharge station the material falling from the upper to the lower run of the belt requires to be moved from a state of rest to the speed of the belt." This has been the experience of others using fixed trippers with by-pass chutes, and at present they are seldom used. A traveling tripper bends the belt less than a series of fixed trippers, avoids all or most of the reloading on the belt, and is likely to cost less than three or four fixed trippers with their chutes. At any tripper, the material must be lifted from 2 to 5 feet in order to effect a discharge. A traveling tripper does this once, but a series of fixed trippers does it oftener. Hence more power is required to drive the conveyor through such a series of fixed trippers.

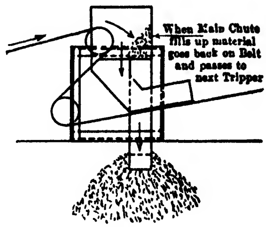


FIG. 9-9. Fixed Tripper with By-pass Chute.

Use of Fixed Trippers. When, however, a belt discharges to a series of small bins, or through hatches in a floor to tanks or chutes beneath the floor, it may not be convenient to use a traveling tripper, because in order to avoid spill on the floor the belt must be empty when

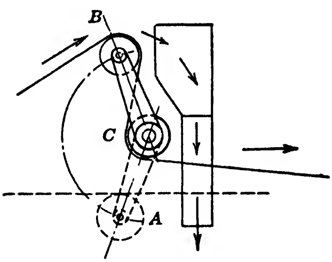


FIG. 9-10. Stationary Tripper with One Pulley Movable to Effect Discharge or Allow Material to Pass By.

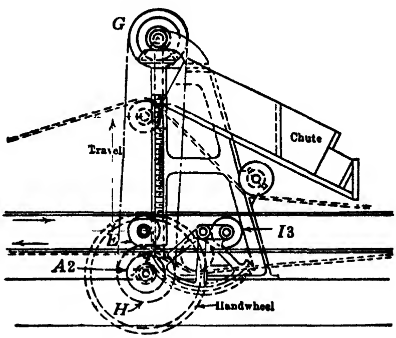


FIG. 9-11. Cookman-Neall Tripper for Grain Belt.

the tripper is moved. In handling grain in American "elevators" it is customary to stop the flow of grain to the belt when the tripper is moved between spouts, but in some manufacturing plants it is impossible to shut off the feed to the belt without costly interruption of the processes. In such cases it may be necessary to use a fixed tripper over each discharge hatch and suffer the added wear on the belt.

Stationary Trippers with Movable Pulleys. Some of the objections to a stationary tripper with fixed pulleys can be avoided by making the upper pulley movable, either as in Fig. 9-10, where the pulley

swings from its idle position at *A* to its operating position at *B* around an axis on which is mounted the other pulley *C*, or as in the Cookman-Neall tripper of 1901, Fig. 9-11. In this device, which was used at the Washington Avenue grain elevator in Philadelphia for a number of years, the power to lift the discharge pulley is taken from the return belt. Turning the hand wheel depresses *I3* and forces the return belt into driving contact with *A2*. A chain running from *H* to *G* drives a pair of vertical screws through reversing clutches to raise or lower *E*.

Traveling Trippers. The earliest trippers used in this country were stationary. Movable trippers came into use after 1870. The first of these were similar to Westmacott's tripper (see Chapter 2). Fig. 9-12 shows one used at Duluth in the early eighties (T. W. Hugo, *Transactions A.S.M.E.*, 1884). It consisted of a pair of cast-iron side frames that carried two pulleys which by means of the hand wheel and worm gear could be rotated from the idle position *AA* to either of the working positions *BB* or *CC*. With the pulleys at *AA*, the loaded belt would pass through the tripper without being acted upon; with the pulleys at *BB*, grain coming from the right would be discharged into the chute; when the belt was reversed to carry grain from the left, the chute was transferred to the other side of the frame and the pulleys were swung to the position *CC*. The frame was pushed by hand to the discharge point, while the belt was slack, and was fastened to the floor by hooks and turnbuckles; the pulleys were raised to the working position and then the conveyor take-ups were tightened.

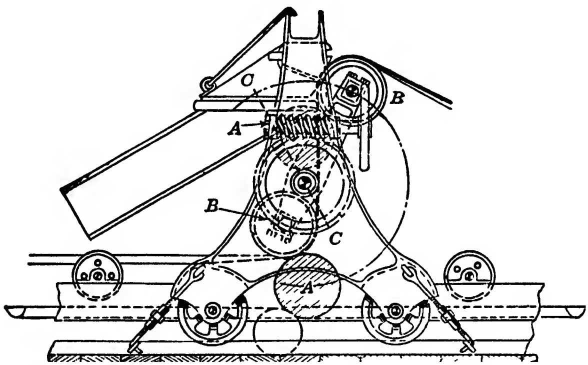


FIG. 9-12. Hand-propelled Tripper with Pulleys Adjustable for Reversible Discharge.

Self-propelled Trippers. William B. Reaney (see Chapter 2) designed the first self-propelled tripper in 1876. It was like Fig. 9-12 in having the pulleys adjustable for discharging a reversible belt, but

it was new in taking power from the conveyor belt to propel the frame backward and forward.*

After a few years of successful use at the Canton Elevator, self-propelled traveling trippers came into general use, and for simplicity in construction they were generally made with pulleys on shafts fixed in

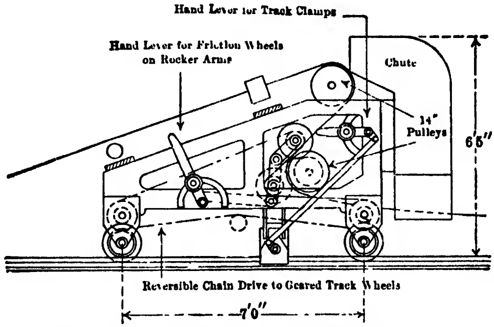


FIG 9-13 Two-pulley Tripper for High-speed Grain Conveyor.

position. In the older hand-propelled trippers it was necessary to swing the pulleys out of position in order to push the frame easily, but that was not necessary with power-driven trippers. At first, separate friction clutches or jaw clutches were used on each pulley shaft to drive chains leading to the track wheels, but later the modern style with paper and iron friction wheels came into use. Fig. 9-13 shows a friction-

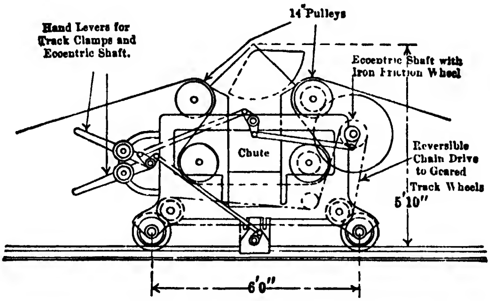


FIG. 9-14. Four-pulley Tripper for Reversible High-speed Grain Conveyor.

driven tripper for high-speed grain belts with geared track-wheels. When a tripper is required to discharge from a reversible belt it is simpler and more convenient to use four pulleys rather than the two movable pulleys of the older style. Fig. 9-14 shows a four-pulley

* Communicated to F. V. Hetzel by Mr. George M. Moulton of Chicago, who with his father, John T. Moulton, built an elevator at Duluth in 1869, the Canton elevator in 1876, and many other important grain elevators.

tripper controlled by two hand levers, one for the friction wheels, one for the rail clamp. The hood over the chute is pivoted and can be tilted to receive the grain from either side. This style of tripper is seldom used for material other than grain; the pulleys are only 12 or 14 inches in diameter, too small for belts thicker than the 4-ply generally used on grain conveyors. A tripper typical of those used for materials like coal or crushed stone is shown in Fig. 9-15. The frame is

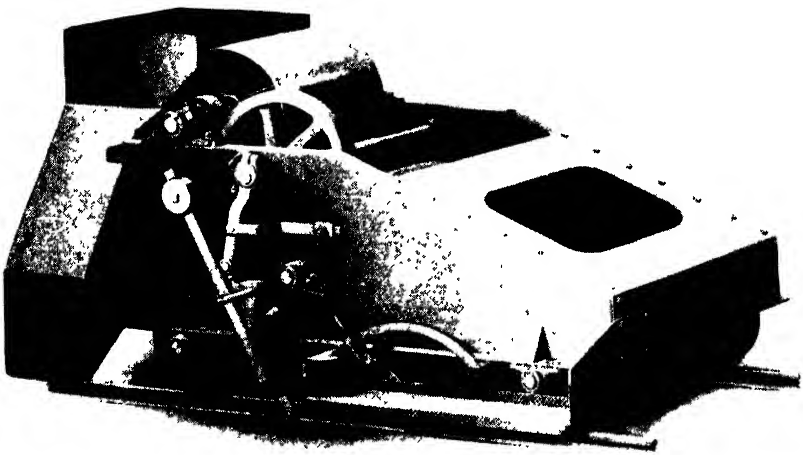


FIG. 9-15. Typical Tripper (1940).

low and compact; the machinery is simple, with the pulley and axle bearings accessible. Friction wheels of compressed paper or rawhide 10 to 12 inches in diameter are keyed on the pulley shafts. The operating shaft is mounted in eccentric sleeve bearings. A lever brings the 20-inch-diameter cast-iron friction wheel on the operating shaft into contact with either of the friction wheels on the pulley shafts. A chain drive connects the operating and axle shafts. With a speed reduction of 6 to 1 between the pulley rim and the rim of the track wheels and assuming a belt speed of 300 feet per minute, traveling speed of the tripper when moving in the same direction as the belt is $\frac{300 - (300 \div 6)}{6}$ or 41.7 feet per minute and $\frac{300 + (300 \div 6)}{6}$ or 58.3 feet per minute when moving in the direction opposite to the belt travel.

Worm-geared Trippers. The original worm-geared tripper patented by Ticknor and Baldwin in 1904 has on one end of the lower pulley

shaft a bevel gear which meshes with two bevel pinions on a shaft lying parallel to the belt and driving each truck axle through a set of worm gears. The bevel pinions on the worm shaft may be clutched to it by the motion of a sliding jaw so as to turn the worm shaft in either direction, or else they are shifted bodily along a key to engage, one or the other, with the driving gear. All the driving mechanism is enclosed, and the appearance is good. The positive drive through gears is objectionable for high-speed belts, but when the belt speeds are under 350 feet per minute, and the tripper pulleys are of good size, the arrangement works well without racking the tripper too much.

Automatic Self-reversing Trippers. One end of a counterweighted lever (Fig. 9-15) engages with a stop located on the track and provides the motion for shifting the operating shaft into engagement with one or the other driving wheels on the pulley shafts. The tripper automatically reverses its direction of travel and will distribute the discharge from the belt over the length of a bin or stock pile without the attention of an operator to set and reset the tripper in position.

Motor-propelled Trippers. When a self-reversing tripper is used on a conveyor that runs two or three shifts per day or when the tripper becomes heavy, owing either to the belt width (36 inches wide and over) or to a special lift or discharge feature, a motor for propelling is mounted on it. The drive from the motor to the axle can be made positive by gears and chains, and reversing the direction of travel is accomplished by reversing the motor. A motor-driven tripper relieves the conveyor belt of the tension required to move the tripper and takes some load off the conveyor motor.

Tripper with Weighing Device. The Uhlig patent, 1,751,898, of 1930, discloses a combination of a tripper with a weighing or measuring machine mounted on the one movable carriage.

Tripper Pulleys. Old grain belt trippers had 12-inch-diameter pulleys as "standard." The custom persists, and some are still built so. They work fairly well with the 4-ply belts common for grain conveyors, but there is no doubt that the belts would last longer if the pulleys were 4 or 5 inches in diameter for each ply of the belt (see page 176), that is, 16 or 20 inches in diameter for 4-ply belts. Grain-conveyor belts do not often fail by cutting or abrasion, but rather from splices pulling apart and from separation of the plies due to failure of the friction rubber. Both these troubles are aggravated and the life of belts shortened by the use of small pulleys in trippers. Large pulleys cost a little more, but they save more than their added cost in the increased life of belts and the avoidance of belt troubles and repairs.

For materials other than grain, the situation is different; the belts

are generally thicker than 4-ply, and the splices hold better; moreover, the life of the belt is often determined by the surface wear rather than the separation of the plies. In such cases it would not always be economy to make pulleys 4 or 5 inches in diameter per ply of belt, especially since belts for heavy work are frequently 6-, 7-, or 8-ply thick. The tripper would become too large and too heavy, and the life of the belt would not be economically prolonged. But though there may be doubt about making tripper pulleys 30 inches in diameter for 6-ply belts, for instance, it is certain that the ratio of diameter to ply should never be less than 3 to 1, that is, 18-inch diameter for a 6-ply belt. Frequently a ratio of 4 to 1 would be better, even in the class of work considered here. Some judgment is needed in selecting the sizes of tripper pulleys; manufacturers' "stock sizes" are often too small for good work.

Extra Length of Belt for Tripper. Because of the bends around the tripper pulleys, some feet must be added to the length of the belt. Makers' catalogs tell how much is needed.

Belt Speed and Tripper Discharge. If a loaded belt traveling 300 feet per minute runs through a tripper geared to travel 42 feet per minute *in* the direction of belt travel, the discharge over the upper pulley during the time of travel makes the same trajectory into the tripper chute as if the tripper stood still and the belt velocity was $300 - 42 = 258$ feet per minute. When the tripper moves *against* the run of the belt at a speed of 58 feet per minute, the discharge is like that of a belt running 358 feet per minute over a pulley in a stationary tripper.

Some trade catalogs give a rule that tripper belts should run at least 300 feet per minute. This "rule" should be used with some judgment. If the belt works in the open and carries material that clings to the belt like wet sand, wet concrete, or loose earth, then 300 feet per minute may be necessary. On the other hand, if capacity or conditions of loading call for a lower speed, or if the material is fragile or easily damaged, it may be better to let those circumstances govern the speed. The reason for the "rule" is that standard trippers are made at the factory to travel about 50 feet per minute on their tracks unless other instructions are given by the purchaser, and to reduce the net effective belt speed by 50 feet per minute when the tripper travels *with* the belt might cause a poor discharge into the tripper chute. But if the tripper is moved only when the belt is empty, or if it travels at a slow speed, or is propelled by hand crank, or if the material is clean and dry and does not cling to the belt, then a belt speed of less than 300 feet per minute may be better than one of 300 feet or more. From an inspection of Figs. 9-1 and 9-2 it is apparent that under some condi-

tions a *net* speed of 200 feet per minute might give a satisfactory discharge into a tripper chute.

If it is important that the tripper should move from one point to another quickly to fill widely separated bins, high speed on the tracks may be necessary, but when the discharge goes into a long bin or into ground storage high speed of travel is not important, and the tripper can be built to travel at a low speed; it will be easier on the tripper and on the belt and tax the motor less.

Tripper chutes are made in various ways; for discharge to one side, to both sides by a split discharge, or alternately, or in such a way as to put the material back on the belt. Ordinary two-way chutes on trippers for coal, stone, ores, etc., are generally fitted with a flap gate (Fig. 9-16) for discharge to either side alternately. When the material

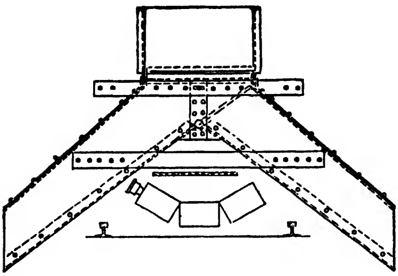


FIG. 9-16. Tripper Chute to Discharge to Either Side.

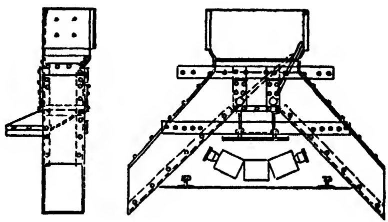


FIG. 9-17. Tripper Chutes to Discharge to Either Side or Back on the Belt.

is put back on the belt for discharge over the end pulley of the conveyor, two flap gates are used (Fig. 9-17). The delivery of material to the belt in this way is not according to the best methods of loading a belt, because the chute is necessarily short and the material does not acquire much velocity in the direction of belt travel.

Tripper chutes are often set with the upper edge at position 2 (Fig. 9-6) so as to save height in the frame and lift of material in passing through the tripper; there are, however, several disadvantages. If the belt is stopped while loaded, starting again at slow speed causes material to fall into the space between the belt and the chute, and if the pieces wedge there and are sharp and angular the belt may be cut or torn. When a chute is placed at position 2, the clearance is generally made small to catch all the material from the belt. It has happened that a blistered belt has caught at the edge of the chute, pulled the tripper loose from the track where it was clamped, and caused a serious accident. It is better to put the upper edge of the chute at a position lower than 2 so as to catch all the spill and yet with

clearance enough to avoid catching torn or blistered places on the belt, or a loose belt-fastener.

Tripper Brushes. Trippers that handle sharp gritty material, especially if wet, should be fitted with rotary cleaning brushes, or else the lower pulley will force sharp particles into the cover of the belt. Since the brush must revolve *against* the travel of the belt, it is usually driven from the upper pulley shaft by a chain which can be shortened when the brush is raised, to compensate for wear.

Belts handling sharp wet sand have been ruined for lack of a brush in the tripper. It must be said, however, that a brush in a tripper needs constant care and attention to keep it in working order. Belt wipers are often used on trippers instead of brushes.

Belts Running Crooked through Tripper. When a belt, instead of running straight through a tripper, runs off to one side so much as to scrape its edges, it is well to look to the gauge of the tripper rails. If there is too much clearance between the rails and the wheel flanges, the tripper frame may pull out of square with the belt. The belt will also run crooked if the ends of the belt are not cut square at the splice or if the tripper frame is not stiff enough to resist distortion.

Trippers should be made so wide that if the belt does run off the pulleys for an inch or two it will not scrape its edges on the side of the discharge chute or on the propelling mechanism of the tripper. Edge rolls are sometimes used to guide the belt in a tripper, but they should be avoided if possible, as they cause wear on the edges of the belt.

Trailers. Large trippers for heavily loaded belts are sometimes made with trailers which are rear extensions of the tripper frame carrying several sets of idlers intended to support the belt as it rises into the tripper (Fig. 9-18). In many cases the trailer is of little or no use, because when the load on the conveyor changes, the belt lifts off the trailer as shown in the figure. The same thing may happen when the tripper approaches the head of the conveyor. The belt tension is greater there, and as a consequence the belt sags less on the lift into the tripper and does not touch the idlers on the trailer.

Length Required for a Tripper. Diagrams of traveling belt trippers given in manufacturers' catalogs show the distance from the chute at the front end to the point at which the straight slope of the belt rising into the tripper meets the line of the horizontal belt. This distance (dimension *A*, Fig. 9-19) is from 14 to 18 feet; but in a horizontal conveyor, the distance *B* from the center of the chute to the point at which the belt begins to lift off the idlers and curve up into the tripper may be 5 to 12 feet more than the distances given in the tables and diagrams. This is shown in Fig. 9-18 where the length is over 25 feet.

Location of First Discharge Point. The point at which the belt begins to lift off the idlers in a horizontal conveyor helps to fix the place of the first discharge through a tripper, because the tripper cannot come any nearer to the loading point without danger of lifting the

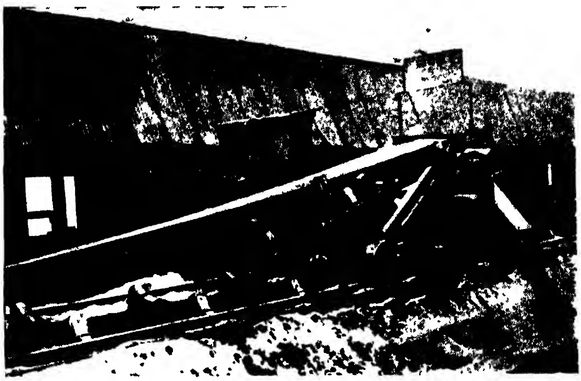


FIG. 9-18. Large Belt Tripper with Trailer Extension.

belt under the loading chute or the skirt boards. If the belt should be lifted by the tripper's coming too close, it will be scraped or cut and perhaps ruined in a few minutes.

It is possible to prevent the tripper from coming too close to the loading chute by using stops clamped to the tripper rails, but there is

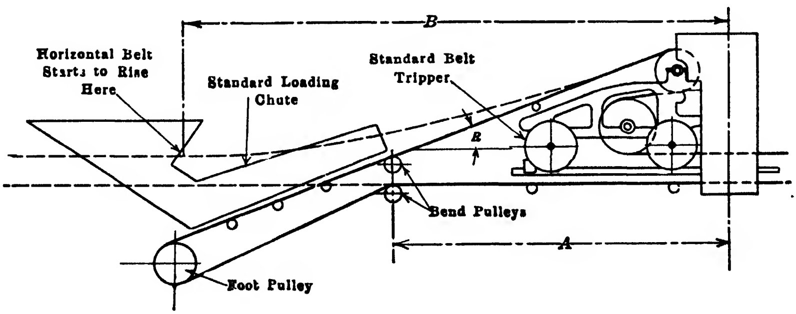


FIG. 9-19. Location of First Point of Discharge through a Tripper with Reference to the Loading Point of a Belt.

always some uncertainty about the point at which the belt will begin to lift, because if it is lightly loaded, or if it is pulled tight by careless use of the conveyor take-ups, or in an effort to get more driving effort at the drive pulley, the belt will lift sooner and span over a greater distance.

To avoid these difficulties, it is customary to depress the foot ends of horizontal conveyors when the first discharge comes close to the loading point. In Fig. 9-19, *B* represents the distance at which the belt begins to lift from a horizontal run, but by humping the conveyor and loading on a short incline the distance is reduced to *A*, and the point of first discharge is brought closer to the loading point by approximately *B*—*A*. Stops should still be used to limit the movement of the tripper (see Fig. 9-20) because, while the depressed end of the conveyor does prevent fluctuations in belt tension from affecting the angle of approach to the tripper, it cannot be depended upon to stop the travel of the

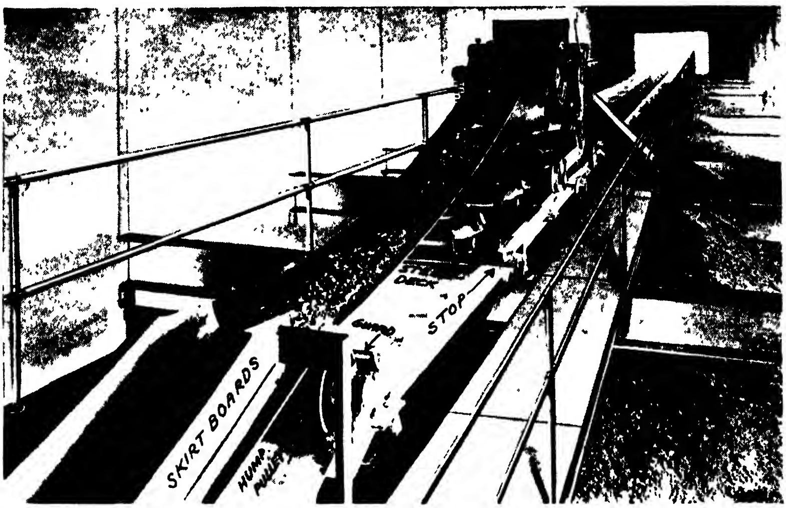


FIG. 9-20. Belt Conveyor with Depressed Loading End, and Stops to Limit Travel of Tripper.

tripper and prevent it from lifting the belt off the idlers on the short incline.

Horizontal conveyors with depressed loading ends are often used in filling short bins where the headroom under a roof is small or where it is important to bring the position of first discharge as close as possible to the loading point. There are, however, some drawbacks to this plan; it introduces another bend in the belt and, what is sometimes more objectionable, loading on an incline. When the short end is inclined at 17° or 18° to match the angle of straight approach to the tripper the material cannot be delivered with proper velocity in the direction of belt travel. At high speeds of belt travel the lumps roll around longer before they acquire belt speed and the skirt boards must be made longer on that account. All this means added wear on the belt.

When the position of first discharge is more than 25 or 30 feet from the end of the skirt boards in a horizontal conveyor it is not generally necessary to depress the loading end, but for shorter distances it is better to depress the end and suffer the added wear on the belt rather than run the risk of spoiling the belt by accidentally lifting it under the loading chute or the skirt boards.

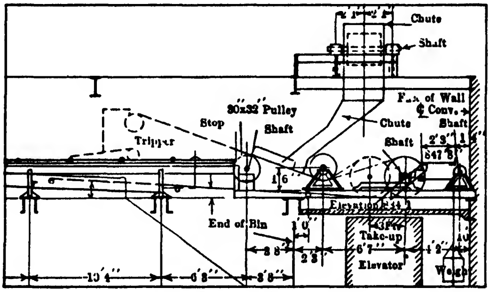


FIG. 9-21. Depressing the Upper Run of a Belt to Permit Loading Close to Travel of Tripper. (Heyl and Patterson, Inc.)

Fig. 9-21 shows the loading end of a 30-inch belt conveyor in a power house where the foot pulley could not be depressed below the level of the conveyor. This arrangement with one depressor pulley accomplishes all that a depressed end does and puts no more bends in the belt. The first discharge in this case comes 14 feet from the end of the bin and 23 feet from the foot pulley.

Devices to Increase the Range of Discharge from a Tripper. In storing soft coal, where it is desirable to limit the depth of pile to reduce the risk of fire, or where it is desired to fill a wide storage building from a central conveyor, or where there are objections to putting the conveyor on a high frame or trestle, it is possible to increase the quantity stored by fitting the tripper with an auxiliary conveyor to carry the discharge off to one or both sides. Fig. 9-22 shows the Blaisdell device patented 1903; the tripper is mounted on a wide-gauge track and fitted with an inclined belt conveyor to pile the material off to one side. In the Moss patent of 1907 the scheme is similar, but the boom which carries the conveyor is

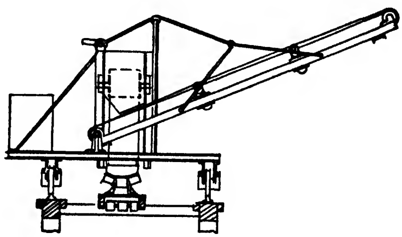


FIG. 9-22. Traveling Auxiliary Belt to Increase the Range of Discharge from a Conveyor.

mounted on a turntable in the tripper frame and is hinged to move up and down. The auxiliary belt will therefore pile material over a considerable area at one setting of the tripper. In another arrangement the tripper carries a reversible belt conveyor on a frame which can be racked in and out at right angles to the main conveyor, and thus discharge material at various distances on each side of the center line.

Extensions of these ideas are shown in other arrangements for distributing materials. The Blaisdell patent of 1902 discloses a main conveyor running along a row of leaching or cyaniding tanks and discharging to a shorter conveyor which spans the width of the tank. This conveyor may carry a tripper for distribution to the tanks and its traveling frame is connected to the tripper of the main conveyor so as to maintain the two conveyors in proper relation to deliver to any point in any of the tanks. In the Dodge device of 1904 a belt conveyor on a low structure discharges to a flight conveyor on a traveling canti-

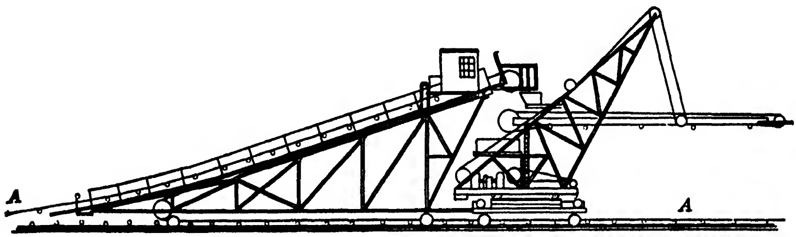


FIG. 9-23. Forming a Pile Alongside a Belt Conveyor by Means of a Suspended Belt Taking the Discharge from the Main Conveyor.

lever frame which also carries the tripper pulleys for the belt. When the flight conveyor is inclined at the angle at which the material naturally piles, a high pile can be formed, the volume depending on the length of the belt conveyor and the length of the flight conveyor.

Stuart Devices. Several patents granted to F. L. Stuart show means to take material from a belt conveyor at ground level and form a storage pile alongside it. Fig. 9-23 (patent 1,331,464 of 1920) shows one scheme. The conveyor *A* runs through a wheeled frame which is in effect a high tripper. A revolving tower mounted on power-driven trucks moves the tripper frame and contains a pivoted boom which carries a belt conveyor that receives material from *A*.

An adaptation of this device is used by the Baltimore and Ohio Railroad at Locust Point, Baltimore, for loading cars. The tripper frame is lower and is fitted with a double-jointed arm, each section of which carries a short belt conveyor capable of 270° angular movement.

In a manner similar to that of the Manierre loader (Fig. 13-2), the conveyor arm can be inserted through the doorway of a box car and the material piled at either end of the car.

The coal-shipping pier of the Baltimore and Ohio Railroad at Curtis Bay, Baltimore, has four 60-inch 12-ply rubber belt conveyors about 1000-foot centers which discharge coal at the head of long inclined loops, which are practically trippers, to reversible shuttle belts carried by traveling towers that load the ships, Fig. 9-24. The shuttle belts can be racked in or out and raised or lowered to suit the height of ship's freeboard and the conditions of loading. The inclined tripper loop is connected to the frame of the shuttle belt and is hinged at its lower end at wharf level, so that the height of the tripper is varied to

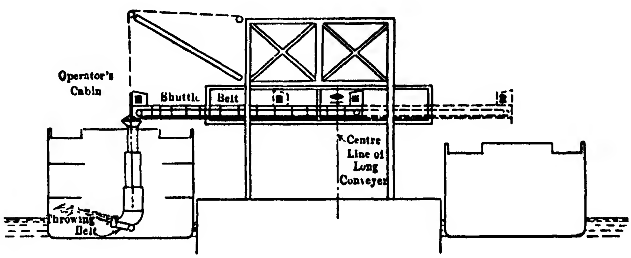


FIG. 9-24. Loading Ships by Means of Shuttle Belts Serving Longitudinal Conveyors on the Wharf.

suit the elevation of the shuttle belt. This and other features of the equipment of this pier are covered by Stuart's patents 1,192,016 of 1916 and 1,241,053 of 1917.

At the coal pier of the Gulf, Florida and Alabama Railroad at Pensacola, Florida, a 42-inch belt carrying 600 tons run-of-mine coal per hour at 475 feet per minute loads ships through a chain and bucket elevator 75 feet high that discharges into adjustable chutes. The elevator is mounted in a power-driven traveling tower which, in addition, carries the loading chutes and a pair of pulleys which act as a tripper. As the tower travels on the pier, the coal on the belt is discharged into the foot of the elevator.

Shuttle Conveyors. In some places where the discharge is spread over a comparatively short distance, or where it is not convenient to use a tripper, it is possible to vary the point of discharge by using a belt conveyor in a portable frame, so that the conveyor receives material at various points along its length, but always discharges over the end pulley. From the fact that they move back and forth, these machines are called shuttle conveyors. In the simple form shown in

Fig. 9-25 the shuttle belt is mounted in a short frame pushed by hand so that the discharge over the end pulley will fall into bins 1, 2, or 3, while a by-pass chute from the head of the elevator discharges into 4. This device does what might otherwise be done with a fixed conveyor

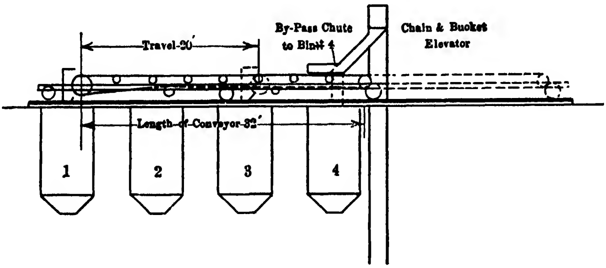


FIG. 9-25. Simple Form of Shuttle Belt Conveyor Serving Three Bins.

with a depressed loading end and a traveling tripper or a series of fixed trippers; in some places it is better than any of these arrangements.

Where the material is to be delivered on each side of the feed point, the belt is made reversible in direction, and it is loaded through a two-

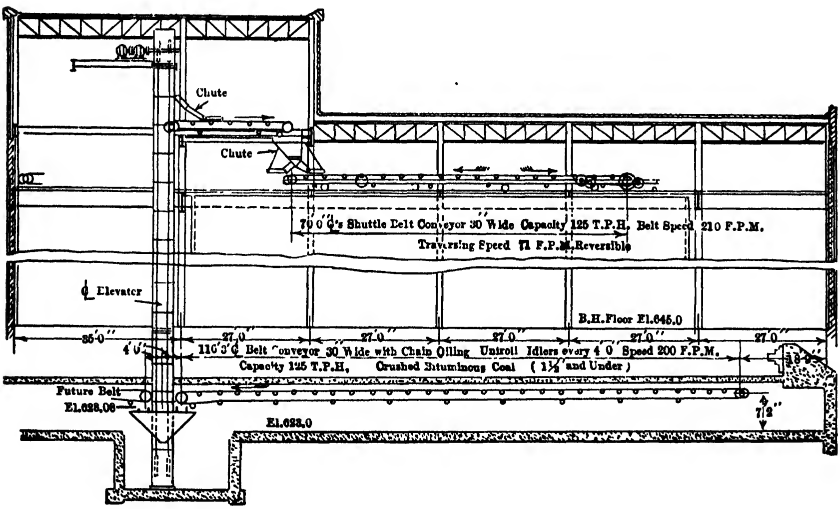


FIG. 9-26. Long Boiler House Served by Shuttle Conveyor. (Link-Belt Company.)

way chute with a flap gate in it. In this way it is possible for a shuttle conveyor of length L to discharge over a length of pile or bin nearly equal to $2L$.

The original shuttle conveyor patented by Bartlett and Overstrom

in 1899 was driven and propelled by a rope drive; as now made, the conveyor is generally driven by electric motor. When the frame is short, or seldom changed in position, the machine is moved by hand, but large frames are more conveniently propelled by a separate motor or by a rope pulled by a winch-head mounted on the machine.

In Fig. 9-24 the shuttle frame is racked in and out by power. The reversible belt conveyor in it receives from another conveyor running the length of the pier, and it discharges material to ships lying on either side of the pier. All the motions of frame and conveyor are controlled by an operator in the cabin on the outboard end of the frame.

Fig. 9-26 shows a boiler house served by an elevator which stands at what will be the middle of the house when the other half is built. A shuttle conveyor 70 feet long serves the present bin which is 104 feet long. When the bin is doubled in length, a short conveyor similar to the present one will run to the left of the elevator, and from the end of it the present shuttle conveyor will take coal for delivery to the new bin. In this way a 70-foot shuttle conveyor will serve 208 feet of bin.

CHAPTER 10

PROTECTING AND CLEANING THE BELT

Material Adhering to the Belt. When coal, coke, clay, and ores are handled, particles often cling to the belt after passing the discharge point and then fall off on the return run on meeting the idler pulleys. If the return idlers are close to a floor or located near the framing of a bent, as in Fig. 10-1, dirt piles up so as to prevent the pulley from turning; the belt may then wear a hole in the rim of the pulley (see Fig. 2-25) and may be cut and perhaps ruined. This is more likely to happen if the conveyor is enclosed in a boxlike housing or if the return idlers are located low with respect to the footwalk, where they cannot be easily seen and reached. It often happens, also, that the return run is hidden from view by the protective deck. In Fig. 10-2 the return run

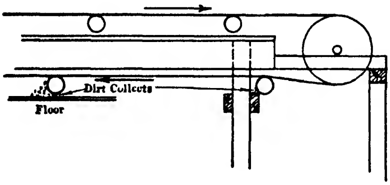


Fig. 10-1. Spill of Dirt on Return Run.

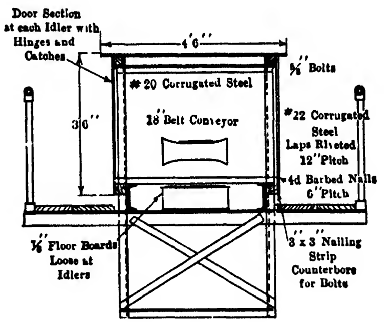


Fig. 10-2. Steel Bridge and Enclosure for Belt Conveyor. Housing Open on the Bottom.

is hidden by the deck, but the gallery is open under the return belt to let dirt fall away. If the floor cannot be omitted, there should be plenty of space below the return belt so that the spill can fall clear of the idlers and be easily seen and removed.

For the same reason, it is well to set return idlers in relation to the framing of bents and the travel of belt so that the dirt falls clear of the bent, as if the belt shown in Fig. 10-1 ran in the opposite direction.

Cleaning Devices. Often the dribble of material along the return

run of a belt conveyor is not serious, but sometimes it is so objectionable that a cleaning device is required near the head pulley to remove clinging particles from the belt. It is also necessary to clean the belt when snub pulleys or bend pulleys or tandem-drive pulleys make contact with the dirty side of the belt on the return run; it not only prevents particles from being forced into the belt by the pulleys, but it also prevents material from accumulating on the rims of the pulleys and forming crusts there which cause the belt to run crooked, or perhaps injure it.

Stationary brushes are not a success for cleaning belts; they fill up with dirt and fine stuff and soon become useless. Air blasts have been used, but they require air under pressure, and the cost of operation is great. Strips of belting set diagonally against the under side of the return belt have been satisfactory on some belts handling coal.

Fig. 10–3. Revolving Brush for Cleaning the Belt.

Revolving Brushes. A revolving brush (Fig. 10–3) consists of bunches of rattan or fibrous splints glued into holes drilled in a wooden cylinder. Cylinders with rubber spirals are also used. The rubber spirals wear longer than the fiber brushes and when wet materials are handled do not clog so easily. Fig. 10-5 shows one with its drive; it must be set so that, when the loaded conveyor is started at slow speed, the material falling vertically from the head pulley will not hit the brush. A chute to collect the scatter and spill back of the brush is not always necessary and should be avoided if possible, but when it is used, the angle must be steep enough to let damp or sluggish material flow readily and there should be room enough between brush and chute to avoid clogging. Fig. 10-4 shows a 24-inch belt discharging crushed coal to a grab-bucket pit. In its first position the chute clogged; when the brush was moved forward and the chute was made steeper it worked well.

Tripper Brushes. Trippers that handle sharp, wet material like sand or crushed ores should have a brush or wiper just ahead of the lower pulley to prevent sharp particles from being forced into the belt or its cover on the reverse bend.

Speed of Brushes. To be effective, a brush must work *against* the travel of the belt and at a speed sufficient to throw the fine stuff out of the bristles and keep the brush clean. Some brushes which do not work well have the bristles or bunches of splints set too close together or they do not revolve fast enough.

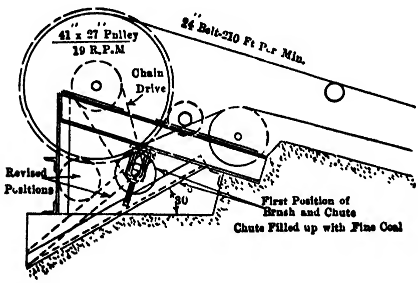


FIG. 10-4. Right and Wrong Positions of Cleaning Brush and Drip Chute.

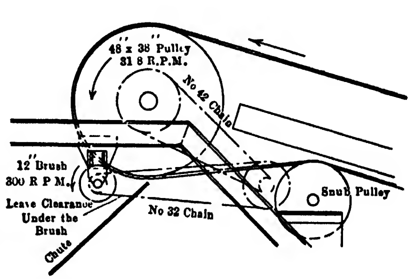


FIG. 10-5. Head of Belt Conveyor and Drive for Cleaning Brush.

The speeds at the tips of the bristles, for brushes 8 to 12 inches in diameter, should be:

Dry materials	800 to 1000 feet per minute.
Damp materials	1000 to 1200 feet per minute.
Wet and sticky materials	1200 to 1500 feet per minute.

The brush should be mounted so that it can be adjusted toward the belt to compensate for wear on the tips of the bristles and in such a way that the drive to the brush is not affected (see Fig. 10-4). A revolving brush does proper work only when it is kept in proper adjustment toward the belt. On that account and because at the high speed necessary brushes are often short-lived, some users of belt conveyors do not employ them at all but let the dirt fall away from the belt and then clean it up regularly.

On some important conveyors, the brushes have been driven by small independent motors.

Belt Wipers. Fig. 10-6 shows a counterweighted belt wiper consisting of a hinged bar faced with rubber belting. This is a simple and inexpensive wiper and has proved satisfactory in many installations. In another form on the market, Sinden patent of 1934, the

hinged bar carries a row of pivotally mounted spring steel flats set diagonal to the direction of belt travel to act as belt wipers.

Other Belt Cleaners. Since it is not easy to repair a fiber brush with the materials and labor ordinarily available a number of substitutes have been developed. Ridgway, in 1912, patented a belt beater which consists of a shaft carrying several pivoted arms on which are loosely mounted pipes extending across the belt. Under the action of centrifugal force, the arms assume a radial position and the pipes strike a glancing blow against the belt. It was designed to work under a free stretch of belt, not against a belt in contact with a pulley. It is not in use now.

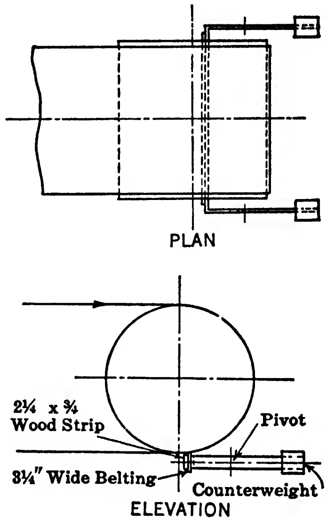


FIG. 10-6. Hinged Belt Wiper.

Fig. 10-7 shows a belt flapper which has some resemblance to Ridgway's device. It is a wood cylinder to which four or five strips of old belt are screwed. It is cheap, easily made, and readily repaired; it is to some extent self-adjusting and will clear itself of fine stuff at speeds less than those given for bristle brushes. The outer edges of the strips are sometimes weighted by steel flats. The Winters patent of 1920 covers a belt cleaner with rubber strips similar to Fig. 10-7 but mounted beneath the return belt on a frame with screw adjustment horizontally and vertically. The vertical adjustment compensates for wear on the edges of the strips, and the other permits the drive from the conveyor head shaft to be kept at the proper length center to center.

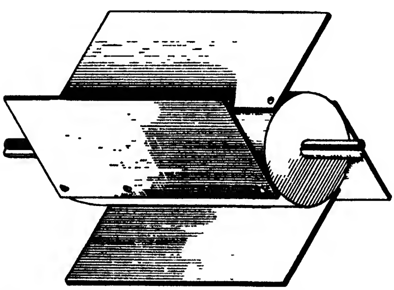


FIG. 10-7. Rotary Flapper to Clean a Belt.

Other belt cleaners have been used experimentally. One is a repair for an ordinary brush; when the rattan bristles wore out, they were replaced by strips of old belt bent in U form and screwed to the wooden cylinder in a helix (spiral) of rapid pitch so that two or three of the edges of the strips were always in contact with the belt. Another scheme consists of discs of old belting 12 inches in diameter strung on a shaft at a slight angle and

nailed to oblique sections of a wood cylinder which support the discs and act as spacers. This is said to work by frictional contact with the belt and to outlast three ordinary rattan brushes, but like many other "kinks" it is likely to work better for the inventor than for anyone else.

The Carr patent of 1920 covers a method to remove, from the return run of a belt conveyor, wet, sticky material like cement mortar. The return belt is kept tight, and, by means of a revolving cam-shaped roller pressing down on it, the belt is caused to whip or snap violently and shake off the adhering particles.

Sprays of water have been used with success to clean belts handling wet concrete.

The idea of cleaning a belt by leading the return run across a wide nozzle which forms the inlet of a vacuum cleaner is disclosed in the

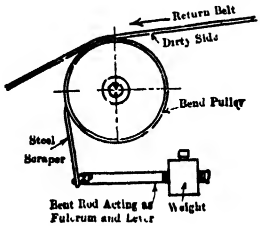


FIG. 10-8 Pulley Rims Kept Clean by Means of a Scraper.

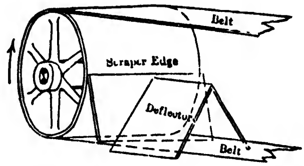


FIG. 10-9. Scraper for Pulley Rim Combined with a Deflector.

Bemis patent of 1918. It is designed to remove dust and dirt from the canvas belts for carrying packages in stores.

Cleaning Pulley Rims. It is sometimes necessary to use steel scrapers on the rims of snub pulleys, tripper pulleys, and deflector pulleys in order to prevent the accumulation of material which might hurt the surface of the belt or cause it to run out of line. Fig. 10-8 shows a scraper fitted to a weighted lever, and Fig. 10-9 illustrates a deflector to shed the scrapings clear of a lower run of belt.

Protective Deck. In order to prevent scattered material from dropping onto the return belt it is usually advisable to cover the space between the conveyor stringers by a floor or deck of plank or light sheet steel. Fig. 9-20, page 236, shows such a deck. It lessens the risk that lumps of material or sticks or tools or similar objects falling on the return belt may be carried between the foot pulley and the belt and perhaps tear it or punch a hole in it. On the other hand, it affords a lodging place for dirt, and on inclined conveyors, lumps falling on the deck may lodge against the idler pulleys in such a way as to form a very effective brake to prevent rotation. In Fig. 9-20 a guard has

been placed across the deck to prevent spill from jamming against the hump pulley.

Inclined decking, Fig. 10-10, is self-cleaning, and on conveyors wider than 24 inches less plate is required to protect the return belt.

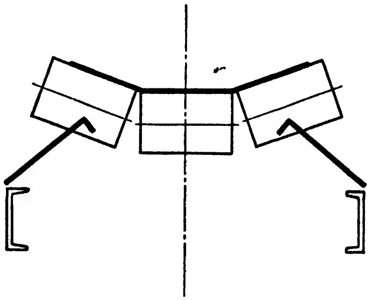


FIG. 10-10. Inclined Decking.

The deck may be omitted from conveyors that carry only fine material and from inclined conveyors where there is a tandem drive or a bend pulley near the lower end of the return run. At such places the lumps can be deflected or thrown off the belt without doing any damage.

When belt conveyors are mounted on elevated frames or bridges, or enclosed in housings, the deck is sometimes objectionable because it covers up the lower run out of sight and hinders or prevents access to the bearings of return idlers. In such cases, if the conveyor belt is plenty wide enough and not likely to spill over the sides, the deck may be omitted and a wiper or plow used over the return belt near the foot to push off any stray piece before it comes to the pulley. Doing this is, of course, choosing between two evils; the

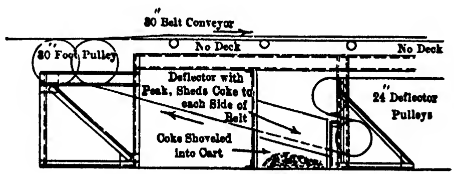


FIG. 10-11. Tripper Device to Remove Lumps from Return Run of Belt. (R. H. Beaumont Co.)

right way is to use a deck and place the conveyor stringers so that the return belt is visible for inspection (see Fig. 10-12).

In some cases where belts are loaded at a number of places along their length as on coke wharves at by-product plants, a deck does not afford complete protection against lumps getting on the return belt and is a place where coke may lodge and do damage. At one plant the return belt was run over two tripper pulleys near the foot wheel. This put two extra bends in the belt but it prevented lumps of coke from being jammed between the belt and the foot pulley, and it gathered the spill at one place convenient for removal (Fig. 10-11). There was no deck over the lower run in this case.

Belt-Conveyor Enclosures. When a belt conveyor runs outdoors throughout all its length, or from one building to another, it may be necessary to cover it for several reasons: (1) To avoid exposure to sunlight, rain, or snow; the material carried on the belt may be damaged if wet, or the belt may carry water into a building or into a bin and create a nuisance. (2) To avoid exposure to wind; a strong wind may blow material off the belt or may lift the belt off the idlers or cause it to run crooked. The simplest protection is merely a roof over the belt, Fig. 10-12, with the sides open. It has the advantage that the carrying run of the belt and the supporting idlers can easily

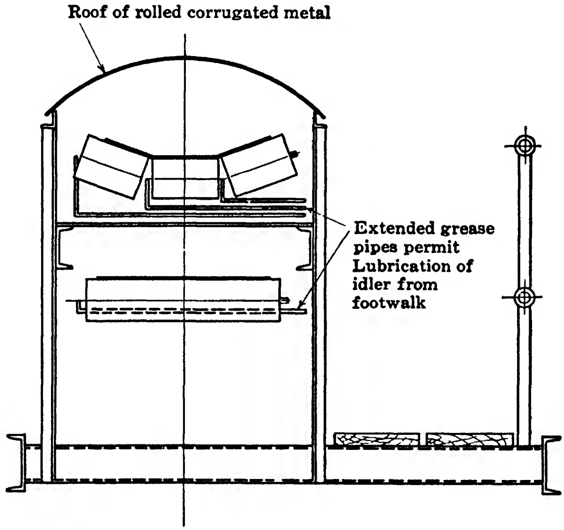


FIG. 10-12. Roof over Conveyor.

be seen. If the belt is not running straight or an idler is not turning, it is more likely to be detected than if the carrying run is completely enclosed. Lubricating the idlers does not present a problem or add to the cost of the installation. The drawbacks are that the material is not completely protected from the weather and in exposed locations wind might cause some difficulties. Fig. 10-13 shows how a belt on a pier was protected from the wind. The belt and material are exposed to the weather, but the conveyor is accessible and the construction permits the travel of a tripper and an intermediate discharge along the belt. When a belt is exposed to strong winds self-aligning idlers are a help in keeping it running straight.

Fig. 10-14 shows a conveyor protected from the weather without

enclosing the footwalk. The curved cover is hinged, made removable, or has doors at the troughing idlers for inspection and lubrication. The return run is above the footwalk where it can be seen.

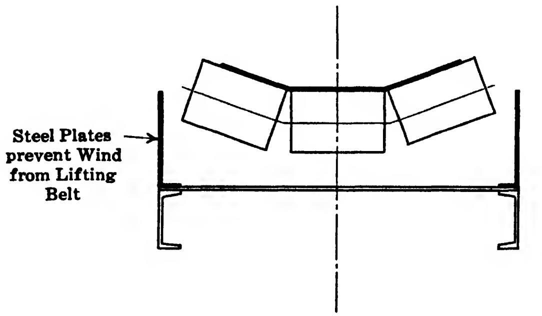


FIG. 10–13. Windbreak for Belt Conveyor.

The return belt is not fully protected from rain, but if the material is not affected by the moisture on the belt no harm is done. This con-

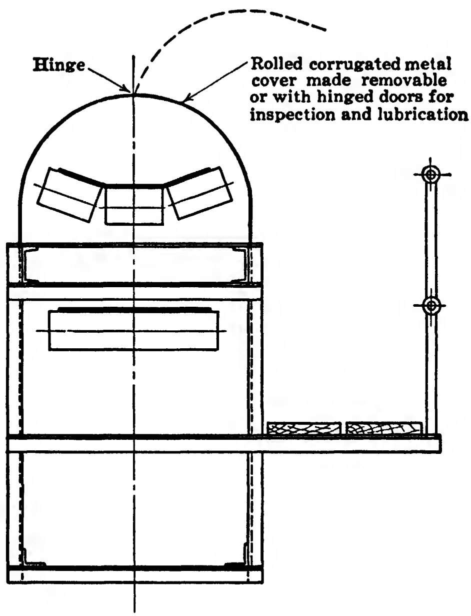


FIG. 10–14. Only Carrying Run of Belt Enclosed.

struction is relatively inexpensive but it has the disadvantage of hiding the carrying run in a tube. If something does not work as expected the trouble may not be discovered until the belt is damaged.

Fig. 10-15 shows an 18-inch belt conveyor enclosed in a through-bridge. The structure was made wide enough for a tripper, but there was a footwalk on one side only and it was not easy to get at the far side of the tripper for lubrication and the necessary inspection and attention. Another defect in the design is that the return run is completely covered. It is hard to get at the bearings, and the belt can be seen only by unbolting and removing sections of the steel deck.

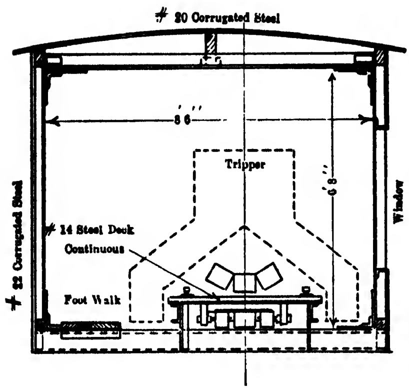


FIG. 10-15. Defective Design of Steel Bridge and Belt Supports.

It is worth while to emphasize the statement that when belts handle coal, coke, and other substances which are regularly or at times moist, some of the fine particles cling to the belt and are dropped off at each idler on the return run, most of them near the discharge end, but often at each idler back to the loading point. This spill is often a serious matter, and if it cannot be allowed to fall freely away at each idler, the supporting structure or enclosure should be designed so that the spill can be removed regularly and easily.

Enclosures which are designed chiefly for architectural effect and neat and trim appearance often cause trouble and expense, because they confine the spill and hinder inspection and cleaning.

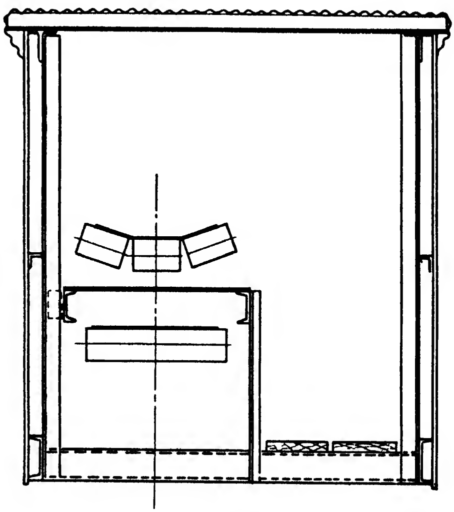


FIG. 10-16. Conveyor Gallery Enclosed on Top and Sides.

Fig. 10-16 shows an enclosed belt-conveyor bridge with a footwalk along one side of the conveyor. Extended grease pipes permit lubrication of all the idler bearings from the walkway. The return run of the belt is located so that it is accessible without removing the decking plates.

CHAPTER 11

WEIGHING AND SEPARATING

Weighing in Transit. Several manufacturers here and abroad supply automatic scales which register the weight of material passing over a conveyor. Fig. 11-1 shows the Merrick conveyor weightometer (Merrick Scale Manufacturing Company, Passaic, N. J.). A short section of the conveyor frame *BA*, independent of the rest, carries

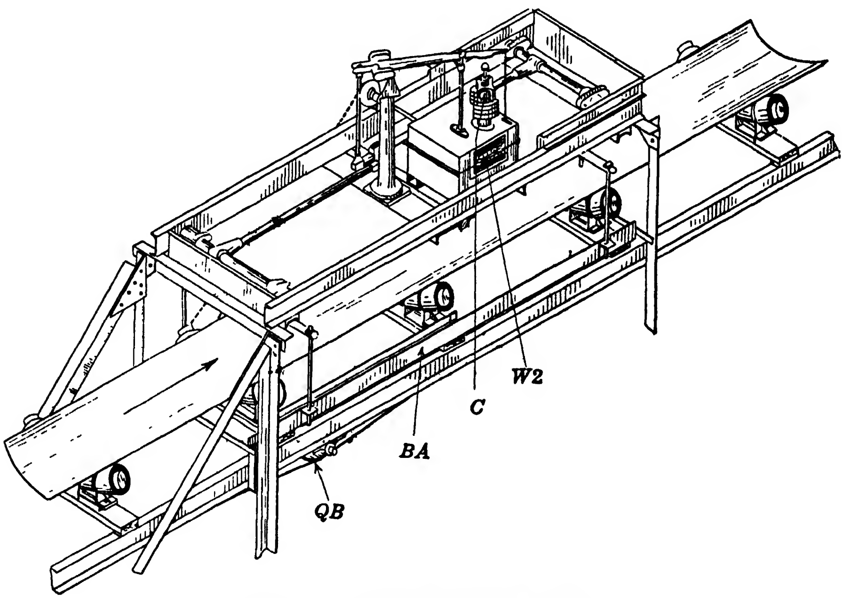


FIG. 11-1. Merrick Weightometer.

three or four idlers which support a few feet of the loaded belt. This is hung from the weighing levers of the scale. The poise or counterbalance on the scale beam dips into a bath of mercury *C*, and, as it rises and falls, it automatically balances the weight of the material on the short length of belt carried by the scale. A device *W*2 called a "totalizer" (Merrick patents 954,869 and 954,870 of 1910) is connected to the scale beam. Its function is to multiply this weight by a factor which depends on the speed of the belt; its operating mecha-

nism is driven from pulley *QB* under the return run of the conveyor belt. A counting machine in the totalizer records the weight of the material carried.

The tension in the conveyor belt as it passes through the weightometer should, for accuracy in the readings, be as nearly constant as possible; hence the machine should be away from the point where the conveyor is driven, and preferably near the loading point. The conveyor should have weighted take-ups rather than screw take-ups, and there should be nothing in the construction or the operation of the conveyor which would tend to lift the belt off the idlers carried by the weightometer. Similar conveyor-weighers are made by Fair-

FIG. 11-2. "Convey-O-Weigh." (Richardson Scale Company.)

banks, Morse & Company, Chicago; Builders Iron Foundry, Providence, R. I., and others.

Proportioning and Batching. Belt conveyors mounted on scales are used as parts of machines for weighing, proportioning, and batching materials. The "convey-o-weigh," of the Richardson Scale Company, shown in Fig. 11-2, uses two conveyors, the upper one as a feeder, the lower one as a weigher. The feeder is stopped and started by the action of mercury switches controlled by the scale mechanism, and the number of weighings is recorded on a counter.

A continuous weigh feeding machine made by Jeffrey Manufacturing Company, Columbus, O., Fig. 11-3, uses an electric vibrating feeder delivering to a weigh belt which is balanced by a scale. The

weigh belt is run at a constant speed. If the weight of material on it is more or less than the amount required to balance the scale the electrical control changes the voltage on the feeder and the rate of feed to the weigh belt is decreased or increased until balance is restored. The material is delivered to the weigh belt at the end farthest from the fulcrum where variations in weight have the greatest effect on the scale and thus increase the sensitiveness of the control.

The poidometer, Schaffer Poidometer Company, Pittsburgh, Pa., combines the continuous feeding and weighing on one short belt conveyor. A section of the belt forms the scale platform, and a regulating gate on the hopper is controlled by electric switches operated by

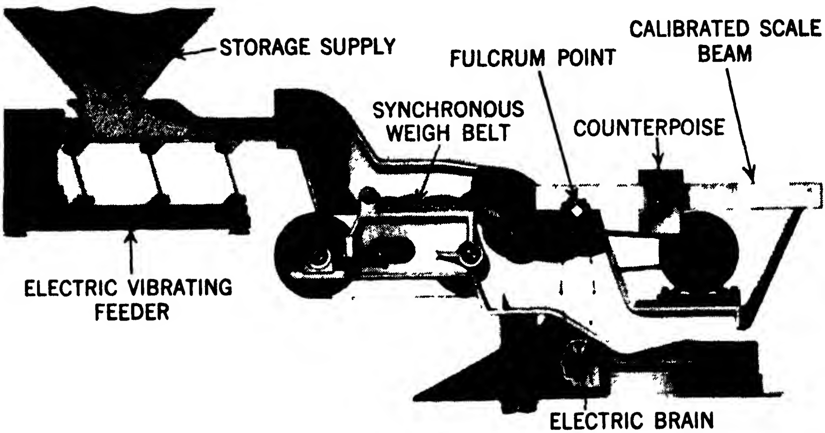


FIG. 11–3. Constant Weight Feeder. (Jeffrey Manufacturing Company.)

the scale beam. If more or less material passes over the belt than the amount balanced by the scale beam the gate is lowered or raised to change the feed to the belt.

The Merrick "feed-o-weight" is similar to the poidometer in arrangement and operation, except that the regulating gate, while controlled by the scale beam, is power operated.

Magnetic Separation on the Belt Conveyor. It is sometimes necessary to separate iron or other magnetic material from the mass carried on a belt. The purpose may be: (1) to protect crushing or grinding machinery against breakage from stray pieces of iron; (2) to separate steel borings from brass chips; (3) to recover usable iron from sand or foundry refuse; (4) to concentrate iron or nickel ores or minerals that have magnetic properties. Where the iron is in stray pieces and small in size and amount, it is sometimes possible to lift

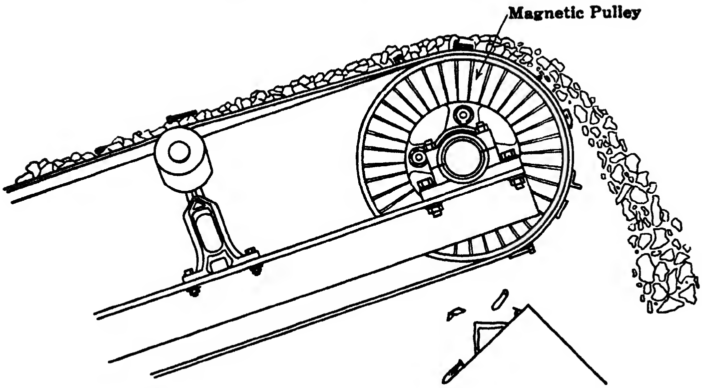


FIG. 11-4. Magnetic Pulley as Head Wheel of Conveyor.

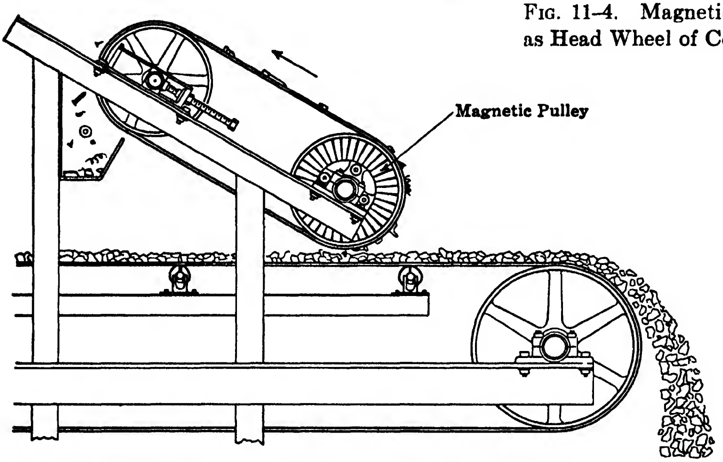


FIG. 11-5. Magnetic Pulley Placed above Conveyor Belt.

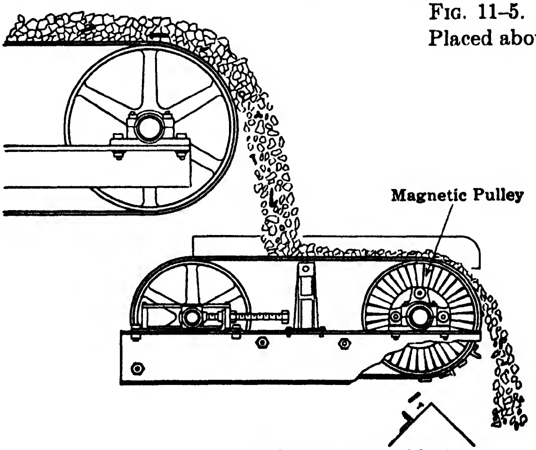


FIG. 11-6. Conveyor Discharging to Wide Separator Belt.

FIGS. 11-4, 11-5, 11-6. Applications of Magnetic Pulleys.

it out of the mass on the belt and hold it suspended by means of a lifting magnet hung over the belt, but if pieces are large and must be got rid of as fast as they are removed from the belt, it is better to use a magnetic pulley. In Fig. 11-4 the magnetic pulley is the end pulley of the conveyor. It attracts pieces of iron, delays their discharge, and drops them into a separate chute. If the position of this chute should interfere with the main chute or prevent the use of a cleaning brush, the magnetic pulley may be placed over the conveyor belt with a separate belt to carry the iron to a chute in some convenient place, Fig. 11-5. If the conveyor belt runs too fast or if the depth of load on it is too great, a magnetic head pulley may not be able to make a proper separation. Under such circumstances it is advisable to spread the conveyed material into a thinner layer on a short independent separating belt, Fig. 11-6, and run it slow. This plan gives the magnetic pulley a better chance to act on the iron or other magnetizable material, and there are advantages in having the work of separating distinct from that of conveying.

Magnetic pulleys are made in sizes up to 30 inches diameter and 48 inches face by Dings Magnetic Separator Company and Stearns Magnetic Manufacturing Company, both in Milwaukee, Wis.

Picking or Sorting Belts. When belts are used to expose ores or minerals to the inspection of men or boys who remove waste material from them the travel must be slow, generally not over 50 feet per minute and even less if the material is lumpy. The bed of material should be thin and wide so that all pieces can be seen and can be turned over without too much exertion. A belt wider than 48 inches makes the pickers (boys especially) reach too far; they cannot work so well, and the sorting is not so efficient as when the belt is 36 inches wide.

Idlers for Picking Belts. Standard three-pulley idlers are not generally used for this work; they crowd the material toward the center of the belt. It is better that the belt should run nearly flat with the material spread out to near the edges; at the slow speed there is not much tendency for the pieces to roll off. Three-pulley idlers, page 102, with a broad-face center pulley are listed for this work by several manufacturers; they spread the material out better than standard idlers. Flat rolls with occasional concentrators (page 100) can also be used.

The number of men required to sort or pick depends upon the size and weight of the pieces and the amount of material to be removed from the belt; it can be determined only by a test under operating conditions. Peele's "Mining Engineers' Handbook," first edition page

1651, says that the weight per hour picked off a belt in 1-inch pieces weighing 1/4 pound is about one-fifth the weight of the material picked off in 6-inch lumps weighing 58 pounds and that the maximum is attained if the pieces are about 3 to 7 pounds.

The length of a picking belt is determined by the number of pickers and by the space alongside allowed for each one; 30 to 40 inches is usual per man on each side of the belt.

The incline should not exceed 10° to 12° to prevent lumps from rolling on the belt and hurting men's hands. Moreover, men cannot stand comfortably on an incline; if the belt is on a slope it may be better to build the working platform as a series of steps; for a 10° slope, a 36-inch tread has a 6½-inch rise.

CHAPTER 12

PACKAGE CONVEYORS

Package Conveying. The belt is generally the cheapest and best conveyor for packages of light weight, papers, books, wrapped goods, sacks, bags, and for boxes that are not too heavy. The belts are always run flat.

Belts for Package Conveyors. Since the work is usually dry and indoors, and the material not harmful to the surface of the belt, it is not necessary to use a belt with a high resistance to abrasion and to the action of water. Many package conveyors that carry light goods use solid-woven belts (see page 64) of a thickness corresponding to 4-ply. If the length is short, the belts may be used just as they are woven, without waterproofing to resist atmospheric moisture; they are cheap, clean, and contain nothing to mark or stain the goods carried. Such belts are very flexible and will bend readily over the 8-inch end pulleys and the 2½-inch idlers generally used in such conveyors. In longer conveyors where the stretch of an untreated belt is objectionable, fabric belts (solid woven or built-up) may be treated with a colorless class 3 compound (see page 63). This is clean and leaves the belt quite flexible. Belts for handling baskets, boxes, heavy parcels, mail bags, express matter, etc., need a density of body and a surface toughness to resist the bumps, blows, and scratches caused by sharp corners, nails, and metal fastenings. They are frequently stitched canvas belts with class 1 impregnation (see page 62), not so flexible as those treated with class 3 compounds, but more resistant to wear. Fig. 12-1 shows one of many such belts used in a Chicago mail-order house. Table 3-K compares two specimens of solid-woven belt and one stitched canvas belt in respect to stretch and ultimate strength.

Fig. 12-1. Stitched Canvas Belt for Assembling Goods on Mail Orders. (Imperial Belting Company.)

Package conveying does not often require a belt with a minimum

amount of stretch or a maximum resistance to moisture; hence rubber belts and balata belts are not often used for this work unless their prices are low enough to compete with solid-woven or stitched belts. When rubber belts are used a cover is seldom needed, experience showing that friction surface belts resist the wear sufficiently well and that cheaper belts are more economical in the long run.

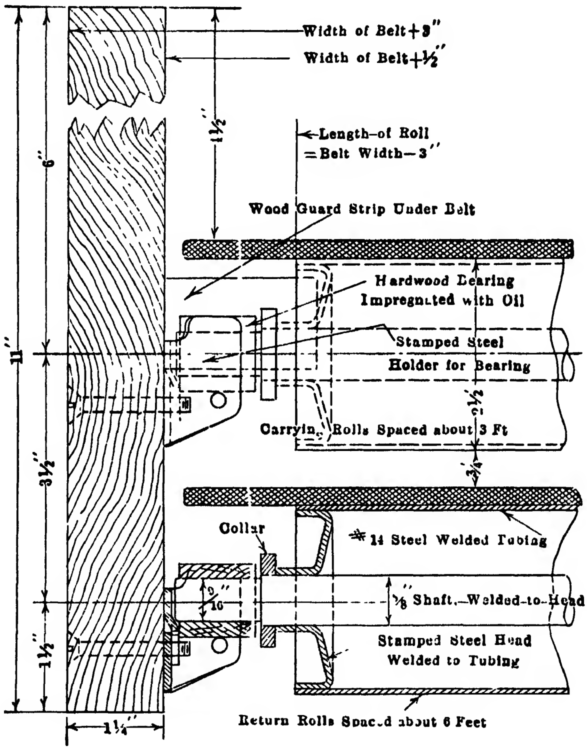


FIG. 12–2. Package Conveyor, Wood Frame, and Oilless Bearings. (Lamson Company.)

Supporting Rolls for Package Conveyors. Wood rolls tend to warp, crack and get out of balance; they are practically obsolete. Steel rolls in sizes 4 inches and larger are generally made of steel pipe or tubing shrunk onto cast-iron heads in which are inserted shafts, either as short stub ends or "gudgeons" cast with the heads or set screwed in place, or as shafts extending through both heads with projection enough for a bearing at each end. For light belts, it is important to have a roll that is light, lively, and accurate in balance; most of them

for store service are made of 2½-inch-diameter steel tubing. If stub-end shafts are used, the cast-iron heads must be of sufficient length to maintain the tube and the two end shafts in permanent alignment. If a through shaft is used, the heads may be much lighter, but the shaft must be perfectly straight.

Most package conveyors are used indoors and in places where neatness and compactness are of some importance. Fig. 12-2 shows a cross section of both runs of belt conveyors from 6 to 48 inches in width, all contained within the depth of an 11-inch plank side. The edges of the belt overhang the ends of the rolls by 1½ inches, but are prevented from sagging too far and dropping goods by a wood guard strip which is continuous for the length of the conveyor except for clearance at the bearings. Fig. 12-3 shows a similar construction used for packages in stores where the conveyor is hung from the ceiling by rods, and where for the sake of appearance it is enclosed on sides, bottoms, and ends with wood construction and wall-board panels. The bearings shown in Fig. 12-2 are hard wood blocks impregnated with a lubricant and held in a yoke which allows them some freedom of movement. Such bearings require no oil; they are sufficient for moderate loads. The bearings shown in Fig. 12-3 are bronze, mounted in cast-iron holders in such a way as to maintain alignment with the roll shafts.

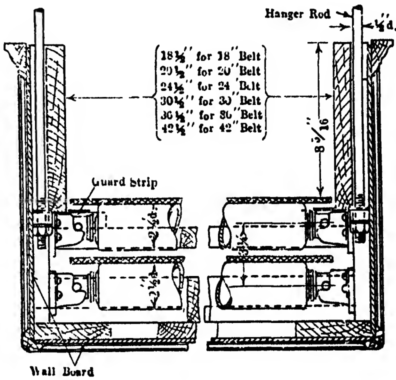


FIG. 12-3. Package Conveyor, Panel Board Enclosure. Bronze Roll Bearings. (Lamson Company.)

Fig. 12-4 illustrates a 42-inch belt conveyor (Link-Belt Company) for carrying mail bags in a postoffice. In the new postoffice in Chicago there are over 700 Lamson belt conveyors for handling mail bags, letters, and packages. The aggregate length of these conveyors is more than 20 miles.

Fig. 12-5 shows typical cross sections of belts for carrying packages.

When a belt slides on a runway or in a steel or wood trough it is customary to use a canvas belt because of the lower coefficient of sliding friction. If stretch is likely to be troublesome, a rubber belt with a "friction" face on the carrying side can be used. Plows will deflect packages more easily from a belt that has no rubber face.

Fig. 12-6 shows a system of belt conveyors in the packing room of a pottery. The ware is carried on pallets or in cartons, which by means of plows or deflectors can be shunted off the belt or transferred from one belt to another. By combining gravity roller conveyors with belt conveyors it is possible to carry packages to or from a number of

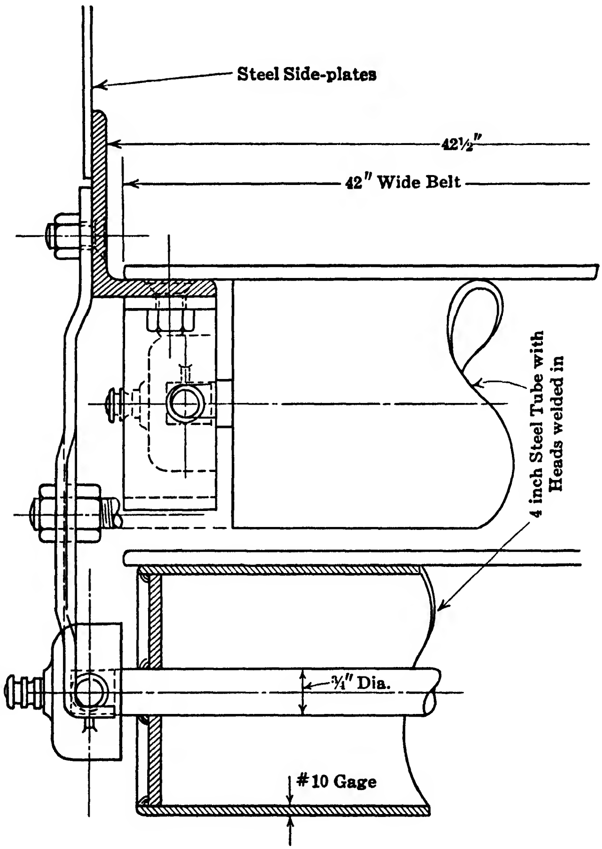


FIG. 12-4. Cross Section of 42-inch Mail-bag Conveyor.

loading and discharge stations and thus serve manufacturing or assembly processes located at a number of points in a factory.

Package Conveyors with Selective Discharge. By a proper combination of guide pins or striking lugs mounted on the packages, and plows set to register with or miss the pins, it is sometimes possible to deflect packages to two or more points along the run of a belt conveyor. Manufacturers show some clever arrangements of this kind.

Rough-top Belts for Package Conveyors. Manufacturers of canvas or solid-woven belts can furnish them with a coating of crepe rubber applied as a cement on the carrying side, and rubber belts can be

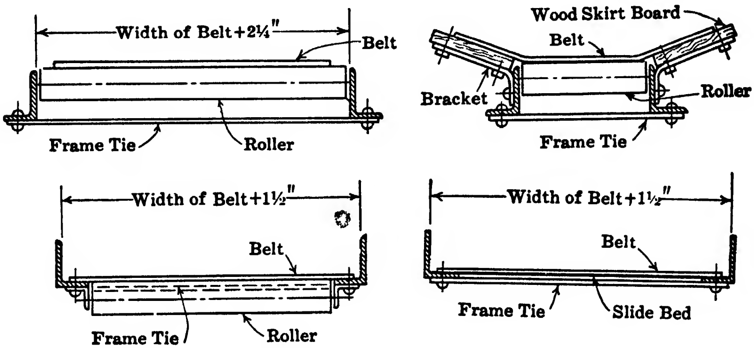


FIG. 12-5. Typical Cross Sections of Package Conveyors. (Logan Company.)

supplied with covers molded with cross corrugations or irregular protuberances in order to make inclined belts carry at an angle steeper

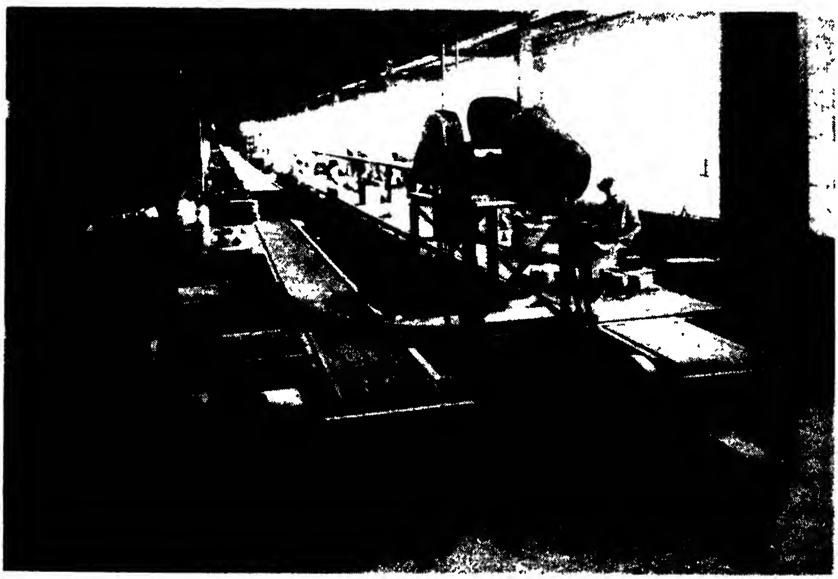


FIG. 12-6. Belt Conveyors in Packing Room of a Pottery. (Lamson Company.)

than ordinary. Some rubber belts of the latter kind sold under the name "claw-top" (patent 1,941,912) are made in three grades of

roughness and according to the maker will carry at the angles listed in Table 12-A. The advantage of the steeper angle is that the conveyor becomes shorter and occupies less floor space. Bags of sugar and similar materials have been carried at angles even steeper than those given in the table. See page 213.

TABLE 12-A

SERVICE RECOMMENDATIONS FOR CLAW-TOP BELT

Material Carried	Type Claw-top Belt	Conservative Angle	Usual Widths, inches	Belt Speeds, feet per minute
Card, paper, or fiber boxes and cartons	Standard	26	16–24	50–100
Light boxed and unboxed materials.	Standard	26	16–24	60–100
Bundles of paper	Standard	26	12–30	80–100
Light material in bags	Standard	28	14–30	60–80
Mail and papers in cloth bags	Medium	30	20–48	100–150
Card, paper, or fiber boxes and cartons	Medium	30	16–24	60–120
Wood and steel beer cases	Medium	28	16–24	50–100
Flour in bags and sacks	Medium	28	16–24	50–80
Cement and lime in bags	Medium	30	16–20	60–100
Cartons and boxes of heavy materials	Medium	28	16–24	50–80
Department store packages	Medium	30	20–36	60–100
Lime and cement in paper bags	Medium	28	16–24	60–100
Trunks, heavy baggage, and boxes	Heavy	26	20–48	40–80

Other Forms of Package Conveyors. Bulletins of manufacturers show package conveyors with the belt supported on one or both runs by oiled wood strips instead of rolls, conveyors with upper run and lower run spaced apart so that they can be loaded with goods on both runs for movement in either direction and conveyors for trays where the belts are carried on rolls set close together.

Capacities of belts in package conveyors are much less, measured in pounds, than the corresponding capacities in handling bulk materials. The service is generally intermittent, the goods are usually placed on by hand, the speed is low—75 to 150 feet per minute—and the width is often determined by the dimensions of the packages rather than by the quantity to be handled. The rate at which the packages can be taken away at the discharge point will sometimes determine the carrying capacity of a belt.

Discharge is generally over the end pulley or by means of scrapers at intermediate points. A typical transfer between two package con-

veyors in a department store is shown in Fig. 12-7. The head pulley is 8 inches in diameter; it is placed quite close to the receiving belt with merely a guard to bridge the few inches of gap. This short connection saves height, lessens the chance of breakage of goods, and prevents an

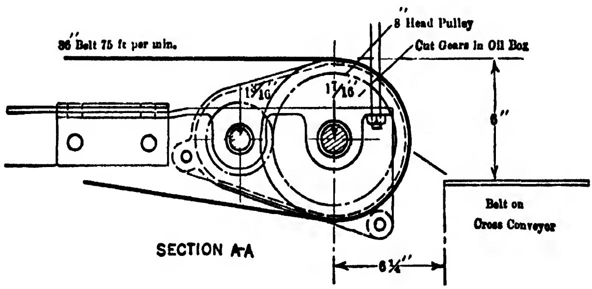


FIG. 12-7. Transfer between Two Package Conveyors. (Link-Belt Company.)

accumulation at the transfer point. In transferring sacks and bulky packages, it is better to place the receiving belt lower, so that the package has more drop and will be taken away promptly; otherwise it may hang between the two conveyors and be rubbed and perhaps torn by the moving belts.

CHAPTER 13

PARTICULAR USES OF BELT CONVEYORS

Portable Belt Conveyors. Nearly all makers of conveying machinery furnish belt conveyors mounted on frames with wheels or crawler treads and issue special catalogs describing them. Such machines are:

1. Wagon or truck loaders, often self-propelled, for picking up stored coal, coke, sand, gravel, etc., and delivering it to trucks or cars, or conversely, taking the material from under drop-bottom cars and

FIG. 13-1. Self-propelled Truck Loader Digging Coal from Storage Pile. (Portable Machinery Company.)

putting it into storage pile or into bins. Fig. 13-1 shows a self-propelled wagon loader taking coal from a storage pile.

2. Bag pilers, that elevate bags or bales to heights at which they are to be piled, or lower them when they are taken from storage, or carry them from storage to cars or boats. Most bag pilers, however, are double-strand chain machines.

3. Portables, to take sand, gravel, or stone from trucks or cars to storage bins or piles or to carry these materials to concrete mixers, or the wet batch from the mixers to the forms in which the concrete is placed.

4. Box-car loaders, in the simplest form a hand-propelled vehicle which carries a hopper that receives dry, loose material from a chute and delivers it to a belt under the hopper which runs at a speed sufficient to throw the material from the doorway of a box car to the end of the car. In another type, shown by the Manierre loader, Fig. 13-2, a belt conveyor is mounted on a steel frame hinged at several points so that it can swing into a box car through the doorway. The conveyor does not run fast. It is adjustable for angle and will pile lump coal or large coke into the ends of a car without breakage, starting with the delivery end close to the floor, and raising as the pile forms.

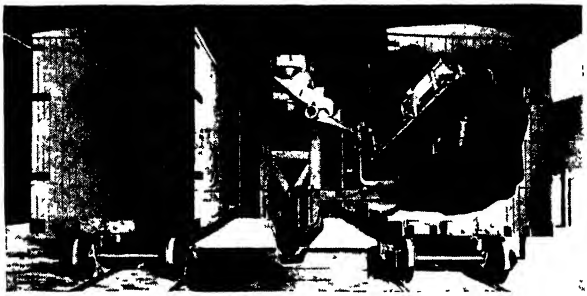


FIG. 13-2. Manierre Loader Receiving from Belt Conveyor and Delivering to Box Car.

5. Ship Loaders. At piers where coal or coke is transferred from cars to sea-going ships, 36- or 48-inch belts are placed at the bottom end of the telescopic loading chute to throw the material away from the ship's hatch to a distance of 30 feet or more and close up under the ship's decks. The belts run fast, 2000 feet per minute or more, and are protected on the carrying surface with sheet-steel reinforcement strips. The life of these belts is short, sometimes measured in hours only. Fig. 13-3 shows a loader with men putting on a new belt.

6. Stackers are inclined conveyors, usually belt conveyors, arranged to take material at the foot end from a car, from another conveyor, or from a power shovel and deliver over the top end to form a storage pile or a spoil bank. Some are mounted on a pivot to swing around and form a storage pile on the arc of a circle. See Fig. 13-4. Some are mounted on frames that run on rails to form long rectangular piles. Fig. 13-5 shows a large stacker mounted on crawler treads; it has a 54-inch belt conveyor on a 190-foot overhang to take material from a power shovel and deliver to a spoil bank at the rate of 800 tons per hour.

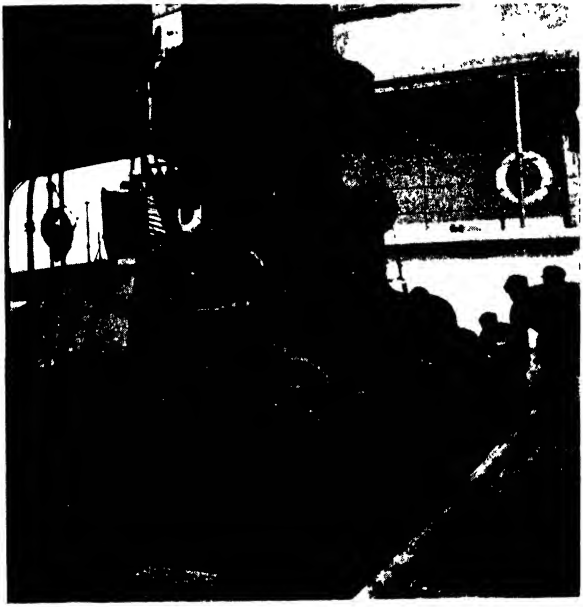


FIG. 13–3 Lower End of Ship-loading Chute Fitted with High-speed Belt for Trimming Coal under Decks

FIG. 13–4. Portable Belt Conveyor Mounted to Pivot and Store Material in a Circular Path. (Barber-Greene Company.)

Portable conveyors in many forms are made by Portable Machinery Company, York, Pa.; Geo. Haiss Manufacturing Company, New York

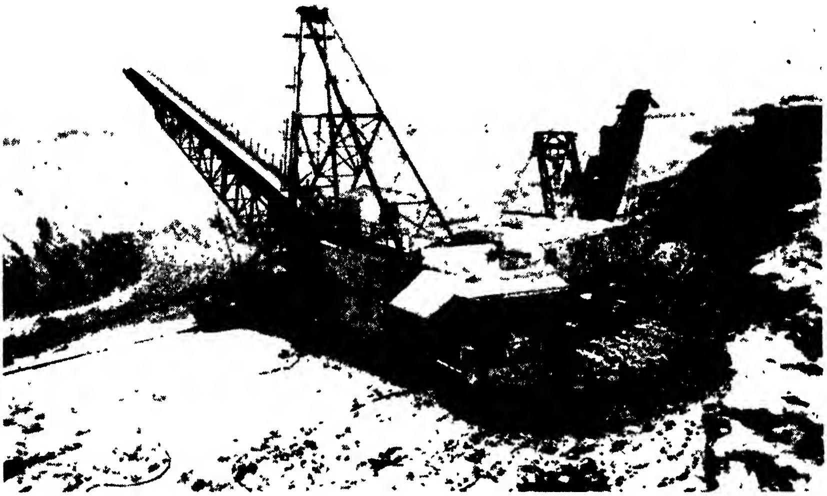


FIG. 13-5. Stacker Receiving from Power Shovel and Delivering to Spoil Bank. (Link-Belt Company.)

City; Barber-Greene Company, Aurora, Ill.; and others. Their illustrated catalogs show various ways of using these machines, either singly, or in combination of several units.

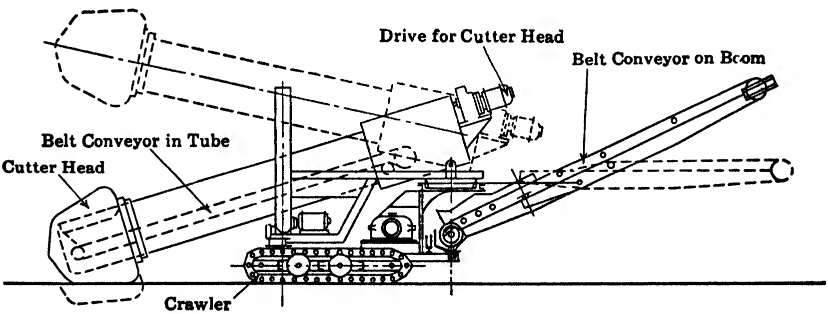


FIG. 13-6. Portable Loader with Cutter-head.

Fig. 13-6 shows a portable digging and loading machine made by Bleichert of Leipzig. A frame mounted on crawler treads carries a tube about 3 feet in diameter and 18 feet long which can be swung

horizontally or vertically. At one end is a revolving cutter-head which digs material from a pile and delivers it to a belt conveyor within the tube. Another belt conveyor mounted on a swingable boom takes the material and can deliver it to a height of 8 feet.

Catalogs of German makers show unloading machines in which a large "shovel wheel" 15 or 20 feet in diameter digs coal from a barge or a pit and delivers it to a belt conveyor mounted on the pivoted boom which carries the wheel. The digging capacity of a 16-foot wheel is said to be 175 tons of coal per hour.

Ready-made Conveyors. Contractors for roads, bridges, dams, or buildings, and persons interested in piling or conveying material outdoors, may save time and money by using "standardized" belt conveyors. These conveyors, made by Barber-Greene Company, Aurora, Ill., and others, are built up in the field of self-contained groups of head machinery, foot machinery, etc., bolted to frames made of steel trusses of certain standard lengths of 6 to 24 feet. The machinery groups and trusses are kept in stock or can be made up quickly, little or no drafting time is required, and erection in the field is said to be quick and easy. These conveyors are made with belts 18, 24, 30, or 36 inches wide and for electric-motor or gasoline-engine drive from 5 to 20 horsepower.

Conveying between Two Belts. The angle at which a belt will carry material up an incline is limited by the tendency of the material to roll back on itself or slip on the belt, but if the material can be prevented from doing that, it can be carried at a steeper angle. It can be done if a second belt travels along with the conveyor belt with slack enough to lie over and confine the material.

This principle has been applied in agricultural implements to elevate straw, etc., and even to carry coal in inclined belts forming part of loading machines, but for bulk materials which consist of a mixture of lumps and fines, the hold of the upper belt is uncertain and some of the material may slip back or fall out. This drawback does not exist to the same degree with packaged goods, and when the pieces carried are uniform in size and not too thick, like newspapers, there is no difficulty in conveying them at angles up to the vertical.

Newspaper Conveyors and Elevators. The original newspaper conveyor of this type (Perkins patent, 1880) is shown in Fig. 13-7. By confining the vertical runs to the two belts *A* and *B* between pulleys set opposite or staggered, the papers are prevented from slipping by the pressure of the belts toward each other. There is one objection to the use of belts for this work, however, when the machines take papers directly from the press, as is usual. The ink is then not quite dry and

will be smeared or smudged if the belts absorb the ink or if in passing over the pulleys there is movement between the belts and the papers. In recent machines of this type, the belts have been replaced by cotton cords wound round with a covering of soft iron wire. The wire touches the papers in small spots only and is not likely to smear the ink. Newspaper conveyors are made by Cutler-Hammer Company, Milwaukee, Wis., and Lamson Company, Syracuse, N. Y.

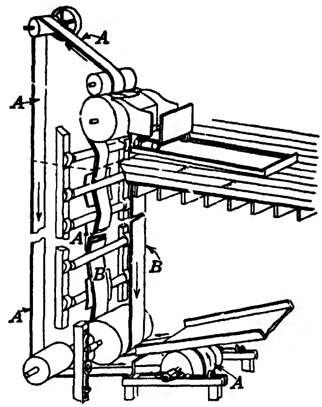


FIG. 13-7. Elevating Newspapers between Two Belts.

The same principle has been used (Lamson Company) in a machine to handle tin crowns or bottle caps. The caps are carried between two belts which are grooved (Fig. 13-8) to such a depth that, when the belts are together, the caps will be squeezed between them. On the vertical run, the belts are guided over a surface slightly convex so as to keep them together and prevent the caps from slipping.

German patent 52,697 to Luther illustrates the same idea applied to bulk materials (Fig. 13-9). It is not in practical use.

The Anderson patent of 1920 proposes to place two short vertical belts under the hopper of a molding machine as a means to draw sand

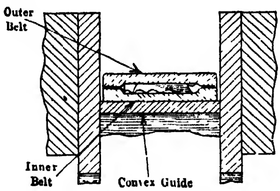


FIG. 13-8. Grooved Belts for Elevating Small Articles. (Cross Section of Vertical Run.) (Lamson Company.)

FIG. 13-9. Conveying Material between Two flanged Belts.

from the hopper and throw it forcibly into a flask or sand mold, so as to pack the sand and lessen the labor of ramming.

Belt Conveying by Rolling Contact. The Hopkins and Fellows device (patented 1904) used in can factories and canneries to carry and elevate round tin cans is shown in Fig. 13-10. Cans rolled down the guide at the foot of the machine rest on a yielding section of track and come in contact with the elevating belt. This is kept under tension

by a weighted pulley and by being deflected from a straight line by the convex curvature of the track on which the cans roll. The motion of the belt rolls the cans up the track until the track ends as shown in the figure, or the track may be curved with a flexible section around the upper pulley to return the cans to the side from which they came.

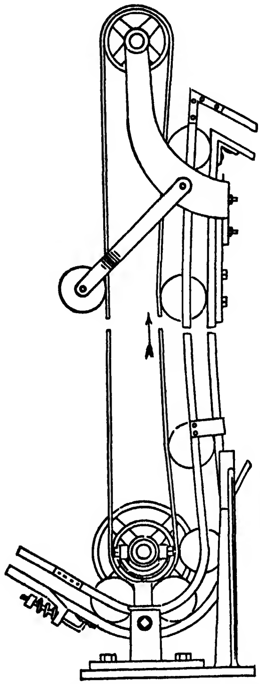


FIG. 13-10. Conveying and Elevating Tin Cans by Rolling Contact.

Long-distance Conveying. To carry large quantities of material over long distances, trains of cars drawn by locomotives may take less power than a belt or a series of belts, and the first cost may be less if conditions favor the use of tracks, single or double. But with better machinery, the power required to drive belt conveyors decreases, and aside from that, a belt system may be installed for the one reason that it costs less for operating labor. For example, preparatory to the design of the conveying system at the Colonial Mines of the H. C. Frick Coke Company,* an estimate was made with the following assumptions: (1) The belt system installed would cost 25 per cent less than a system of haulage by electric locomotives. (2) The cost of depreciation of the belt system would be 15 per cent higher, based on an estimated life of 3 years for belts, 5 years for idlers, 15 years (average) for other machinery, 30 years for steel and concrete. (3) The cost of power for the belts would be 100 per cent higher. (4) The cost of labor and attendance for the belts would be 50 per cent less. All these figures taken together, for a daily output of 8500 tons over a run of 25 years, showed an estimated saving of 3 cents per ton in favor of the belts.

Conveying System at Colonial Mines.† This has 34 apron feeders taking coal from a 1250-ton bin and delivering it to a 60-inch belt which in turn discharges to a series of nineteen 48-inch belt conveyors run at 500 feet per minute and of lengths ranging from 321 feet to 2439 feet. The total lift in the run of 22,930 feet is 357 feet. The troughing idler is shown in Fig. 4-12. The rubber belts are straight

* See *Coal Age*, Dec. 25, 1924.

† T. W. Dawson, before American Institute of Mining and Metallurgical Engineers, February, 1925.

8-ply, 32-ounce duck with $\frac{3}{16}$-inch top cover and $\frac{1}{16}$-inch cover on the pulley side; the joints, over 100 in all, were made in place with an electrically heated, portable vulcanizing press (see page 76). In most cases the conveyors are driven by lagged tandem pulleys placed about 100 feet back from the head pulley and so arranged that the first pulley of the tandem, which does most of the work, drives on the clean side of the belt. All the shafts in the driving machinery, take-ups, and deflector groups are mounted in Hyatt roller bearings.

The stopping and starting of the conveyors in proper sequence are controlled and safeguarded by a complete system of push buttons and switches, and in addition there is a mechanical interlock (Smith patent, 1924) between the head pulley of each conveyor and the foot pulley of the one to which it discharges, to prevent the piling up of coal at these points in starting or stopping the conveyors when loaded. This interlock engages whenever the rear conveyor tends to run faster than the one ahead of it. There are also safety devices that act in case a belt should break or tend to run backward, or if it should slip more than a predetermined amount on one of the pulleys of the tandem drive.

This conveyor system * was started in April, 1924, and is still in successful operation. In its first eleven years it had carried more than 29,000,000 tons. Careful records show that, instead of taking more power than an equivalent electric haulage system, the belts take less, the depreciation is materially less, and the final cost of transferring coal to the river is below the original estimate.

In 1928, the owners of the Colonial belt conveyors installed a similar system 2.9 miles long of 60-inch and 48-inch belt conveyors, at Palmer Dock. These run at 550 feet per minute, carry 1960 tons per hour, and some are about 2000 feet centers.

In 1929 the city of Seattle used 36-inch belts, one of them 1522 feet centers, running at 600 feet per minute, to remove excavated earth at the rate of 900 cubic yards per hour from Denny Hill in the city about a half mile to barges at the waterfront. The conveyors were carried in wooden structures in or over the city streets, ran $23\frac{1}{4}$ hours a day, made no noise, did not interfere with traffic, and did not damage street paving as a fleet of trucks would have done. The belt conveyors were more satisfactory than other systems, trucks, locomotives and cars, and hydraulic sluicing which had been used before 1929.

For a description of the removal of earth by belt conveyors during the building of the Boston Vehicle Tunnel, and the transfer through an air-lock, see *Engineering News-Record,* June 30, 1932.

In building the Fort Peck Dam on the Missouri River, excavated

* Auld and Huttle patent, 1924.

material from four diversion tunnels was carried to spoil banks by two systems of belt conveyors, one tandem of 30-inch belts 3000 feet long, and one of 36-inch belts 2950 feet long. The conveyors at the delivery ends had telescopic sections to feed to the movable piling machines or "stackers" that formed the spoil-banks. For a description see *Compressed Air Magazine,* March, 1935, also *Engineering News-Record,* May 23, 1935.

At the Grand Coulee Dam on the Columbia River, a total of 10,000,000 cubic yards of spoil from the excavations was removed at the rate of 2500 cubic yards per hour by a system of 19 units 60 inches wide, totaling 4648 feet of conveyor, in addition to 1400 feet of feeder belt conveyors. The lift was about 600 feet, belt speed 620 feet per minute (Jeffrey Manufacturing Company). See *Engineering News-Record,* August 1, 1935.

The longest belt conveyor built up to 1940 is one which is 4835 feet center to center at Grand Coulee. Of the total length, the first 2046 feet has a descent of 86 feet, then 1900 feet with a fall of 57 feet, and last 815 feet with a drop of 27 feet. The belt is 48 inches wide, has 8 plies of 32-ounce duck, with a $\frac{1}{8}$-inch top cover and a $\frac{1}{16}$-inch bottom cover; cover stock is 3500 pounds tensile; friction is 20 pounds tensile; all plies have skim coats. The belt carries 2000 tons per hour of stone at 450 feet per minute and is driven by a 125-horsepower motor geared to a single 54-inch lagged pulley snubbed to 210° belt wrap. There is a gravity take-up adjacent to the drive.

The belt weighs 80 tons and was shipped from the factory in eight pieces each about 1200 feet long. These were joined and made endless on the job with an electric vulcanizer.

An alternative design considered at one time for the 4800-foot transfer was to make it by three conveyors of the lengths and slopes stated above and arranged in series. It was known, however, that in a similar installation, it had been found necessary to keep a man at each transfer chute to dislodge chokes and prevent injury to the belts, and it was estimated that the expense of keeping two men on duty 24 hours a day would more than offset any possible saving due to using three shorter conveyors. It was therefore decided to use one conveyor 4835 feet long.

In building the Quabbin Dam for the city of Boston, six 36-inch belt conveyors were used to carry dirt from borrow pits; the total length was about half a mile. (*Engineering News-Record,* May 19, 1938.)

Prior to 1937 the Coplay Cement Company, Coplay, Pa., hauled blasted rock from the quarry to the mill by car. This system has been replaced by three 24-inch belt conveyors, the longest 1264 feet centers,

which carry crushed rock to the mill from a crushing plant on the quarry floor. (See *Rock Products,* August, 1939.)

Fig. 13-11 shows a system of nine 30-inch belt conveyors which in 1937 replaced cars and locomotives at the Spruce Mine of the Oliver Mining Company at Eveleth, Minn. The 30-inch belts carry 750 tons of iron ore per hour a distance of 4400 feet and lift it 385 feet at a belt speed of 500 feet per minute. When locomotives and cars were used, it was necessary to leave some ore unmined to form benches and terraces for the tracks from the mine floor to the surface of the ground. With the conveyor system the terraces are not needed, more of the ore can be mined, and the excavation can be carried deeper without difficulty. (See *Mining Congress Journal,* December, 1938.)

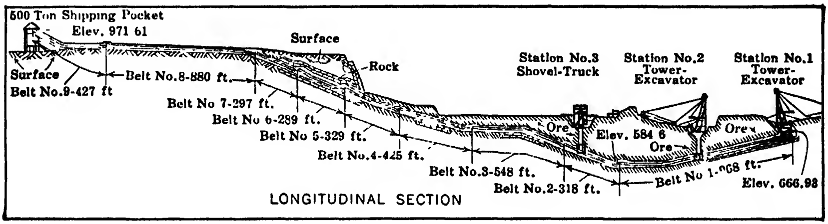


FIG. 13–11. Belt Conveyor System 4400 Feet Long for Iron Ore, Open-pit Mining.

In 1938 at the Roan Antelope Copper Mines in Northern Rhodesia a 1.3 mile tandem of four 42-inch belt conveyors was installed to carry ore at the rate of 850 tons per hour.

Fig. 13-12 shows a system of twenty-six 36-inch belt conveyors in California to carry gravel and sand 9.6 miles to a concrete-mixing plant at the Shasta Dam on the Sacramento River. The conveyors vary in centers from 850 to 3415 feet, and carry 1100 tons per hour at 550-feet-per-minute belt speed. The conveyors start at 490 feet altitude, rise to 1450 feet, then drop down to 650 feet. At the mill which will supply cement for the Shasta Dam, three 36-inch belts making up 1.1 miles in length transport crushed rock 5 inches and under at 500-feet-per-minute belt speed. (Chain Belt Company.)

For a description of the concrete mixing plant at the Friant Dam at Fresno, Calif., in which long belt conveyors are used, see ***Rock Products,*** November, 1940 (Stephens-Adamson Company).

The materials-handling machinery of the Permanente Corporation, San Jose, Calif., using many belt conveyors, is described in ***Rock Products,*** December, 1940.

Underground Belt Conveyors. Underground belt conveyors are coming into wider use as a means of carrying coal away from the work-

ing face in the mine. These conveyors must often work in small space and low headroom, conditions which sometimes dictate a compromise with what is considered good practice in ordinary belt conveyors. To save weight and headroom the diameter of the idlers is made small and the carrying and return runs of the conveyor are brought close together. The return run of the belt is not exposed for inspection, but under the operating conditions inspection is very difficult and uncertain. The length of the conveyor must often be changed as the work progresses. This requirement places a limit on the size and weight of the conveyor sections and the separate parts. The conveyor rests on the

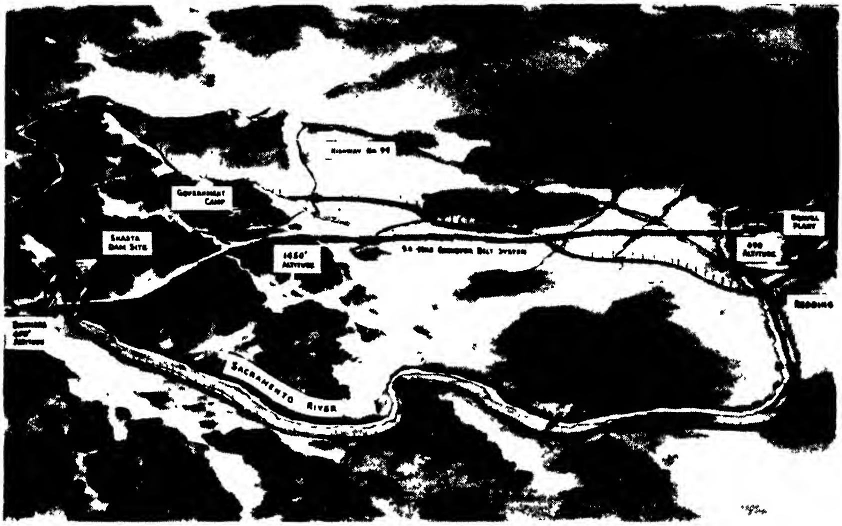


FIG. 13–12. Conveying Sand and Gravel 9.6 Miles to Site of Shasta Dam, California.

mine floor, and the construction must permit the conveyor to follow the undulations that are often present. It is not practical to spend the time to obtain the same accuracy of alignment that is found in ordinary conveyors. It is desirable that the conveyor be reversible so as to carry mine timbers and supplies into the mine when it is not removing coal. Underground conveyors are often required to operate under unfavorable conditions, and their success in England, Germany, and this country has been achieved in the face of many difficulties.

Practice in Great Britain. "Face conveyors" which take the mined coal away from the working face are loaded from the side by hand and have the belts flat or nearly flat. The supporting frames are steel sections about 6 feet long each having a carrying roller and a return

roller. These are steel tubes 2½ or 4 inches in diameter mounted on through shafts and fitted with ball bearings. The carrying rollers are shorter than the width of the belt; the overhang of belt at each side slides on flanged steel plates which form the top of the frame, so that the belt is the moving bottom of a rectangular trough 4 to 6 inches deep. These side plates act as stops for coal tossed or shoveled onto the belt, prevent lumps from falling off, and increase the amount of coal the conveyor can safely carry. Belts are from 16 to 26 inches wide and run at speeds seldom more than 150 feet per minute. As to quality, the best belts for face conveyors are said to be equal to or better than British belts of first grade, see page 56, but after some use the edges wear off until the belt becomes too narrow for further use. Face conveyors are driven by compressed air or electric motors through a set of worm gears and a pair of spur gears. Head and foot pulleys are small, 16 inches in diameter or even less. The head pulley is not lagged; the empty belt is pressed hard against it by a rubber-covered "jockey" or snub pulley, see page 164, backed up by spiral springs. This "mangle-roll" drive can put into the belt a pull much greater than that which is due to belt wrap alone. It is therefore possible to work the belt at unit tensions higher than those recommended for ordinary use. Face conveyors often work under bad conditions, such as wet, dust, mud, bad alignment, and heavy overloads, and these excessive tensions are tolerated. It is considered more important to get the coal out than to spare the belt.

Some face conveyors in British mines are of great length. Some are mentioned in the bibliography, page 276. The catalog of the Mining Engineering Company, Worcester, England, refers to one 900 feet centers, down grade 1 in 25, with a 26-inch belt driven by a 20-horsepower motor through a 16-inch head drum and a 10-inch "jockey pulley."

Where it is not necessary to load from the side and where fairly good alignment can be maintained, conveyors in British mines use troughed belts. Idlers have three rolls, the horizontal one 50 per cent longer than the side rolls, which are set at 30° angle and overhang the supporting brackets. Rolls are of cast iron, 4-inch diameter for belts up to 26-inch, 5-inch diameter for wider belts. The hubs are fitted with ball bearings which run on 1-inch steel tubes; lubrication is by small screw-top grease cups set within the diameter and length of the inclined rolls so that nothing projects beyond them. Belt speeds are seldom more than 250 feet per minute.

Belt conveyors carrying on the lower run have been used in a few places. Loading is easy, and both runs of the belt are accessible for

inspection and repair. The discharge of coal, however, is not simple or easy.

For information on the methods and practice of conveyor mining in Great Britain see the following articles in *Mechanical Handling:*

Developments of conveying and loading adopted by Mavor & Coulson, Ltd., Glasgow, Scotland. Face, gate and gate loaders, November, 1934, pages 327-331.

Comparison of the relative operating cost of flat and troughed gate belt conveyors, H. Maskrey, December, 1934, pages 345-347.

Belt conveyor manufactured by British Jeffrey-Diamond, Ltd., Semi-trough type, December, 1934, pages 353-355.

Face conveyor manufactured by Richard Sutcliffe, Ltd., Semi-trough type, January, 1935, pages 23-25.

Face conveyor manufactured by the Mining Engineering Company, Ltd., February, 1935, pages 55-58.

Gate belt conveyors by the Mining Engineering Company, Ltd., March, 1935, pages 91-94.

Two types of face conveyors patented by Hugh Wood & Company, Ltd., April, 1935, pages 119-122.

Gate belt conveyor patented by Hugh Wood & Company, Ltd., Newcastle on Tyne, May, 1935, page 155.

Troughed belt loader, Mavor & Coulson, July, 1935, pages 217-218.

Gate conveyor extended from 90 feet to 2796 feet in about 2 years, the Mining Engineering Company, Ltd., June, 1937, page 162.

Long underground conveyors, June, 1938, pages 186-187. (Meco troughed conveyor, 24 inches wide. Original length 240 feet at start of operations, October, 1934. Length in June, 1938, 3000 feet, ultimate extension to 3420 feet. Capacity 500 tons per shift. Original belt was in use after handling 500,000 tons. No reportable accidents. Also another installation of a level conveyor, 26 inches wide, length in June, 1938, 700 feet, ultimate extension to 2270 feet. Capacity 280-300 tons per shift.)

Practice in the United States. The Claghorn patent 1,817,348 (1931) describes a flexible and extensible belt conveyor adapted to rest upon and conform to the mine floor, with the drive located at the delivery end, and composed of sections with means to anchor and hold the sections in place. As developed by the Goodman Manufacturing Company, Chicago, Ill., the foot end may be moved back gradually by uncovering a telescopic section. When this section is fully uncovered a new intermediate section is inserted in the conveyor line and the telescopic section is moved beneath the foot section, ready for the next movement of the foot end. A long take-up movement (8 feet 6 inches) is provided in the drive section to furnish belt storage so that the foot end can be moved gradually. The conveyor was designed for room work with a flat belt and side plates.

Experience has shown that flat belts are to be preferred:

1. Where the conveyor is of moderate length and capacity.
2. Where the conveyor length must be changed often.
3. Where the position of the conveyor must be changed often.
4. For conveyors that are loaded at a number of places.
5. For conveyors where the grade is decidedly in favor of the load.
6. Where the mine bottom is soft or subject to heaves.

On the other hand, troughed belts are best for large capacity, long length, and where the grade is against the load.*

The Levin patent 2,168,622 (1939) developed by the Jeffrey Manufacturing Company, Columbus, Ohio, provides for the power unit mounted as a separate piece at the side of the conveyor at any convenient place along the length of the conveyor. A telescopic section and a loop for belt storage permits gradual extension of the foot shaft. The conveyor is reversible.

In the Kendall patent 2,197,187 (1939), as developed by the Stephens Adamson Manufacturing Company, Aurora, Ill., the separate 8-foot sections are made of pipe, each piece of which can be easily handled. The conveyor is made reversible, and the idlers are pivoted near the base so that they will tilt in the direction of the belt travel and assist in guiding the belt. (See page 110.) The tandem drive can be located at the most convenient point along the return run.

Belt conveyors for underground work are made by Robins Conveying Belt Company, Passaic, N. J., La Del Manufacturing Company, New Philadelphia, Ohio, Link-Belt Company, Chicago, and others.

Additional information on the practice and thought in this country can be found in the following articles in the *Mining Congress Journal:*

Conveyor system, Barnesboro Mine, Vol. 21, pages 27-29, April, 1935.
Underground conveyor equipment, M. J. Andrada, Vol. 21, page 24, July, 1935.
Conveyor mining, C. R. Claghorn, Vol. 21, page 45, July, 1935.
Harry Taylor mine of the Penn Anthracite Mining Co., First all-conveyor mine, Vol. 21, pages 21-22, October, 1935.
Room belts, C. R. Claghorn, Vol. 23, July, 1937, pages 40-41.

Two articles in *Coal Age* give information on this subject:

Conveyor mining in southern West Virginia, Virginia, northern Tennessee, and eastern Kentucky, Vol. 39, September, 1934, page 337.
Mining methods and mechanization from 1911 to 1936, Vol. 41, October, 1936; Conveyor mining grows, pages 402-405; Milestones in mechanical loading, tabulation of major steps in loading, entry-driving, conveying, and scraping, pages 405-408.

* C. R. Claghorn in *Coal Mining.*

Practice in Germany. Belt conveyors are gradually displacing shaking or jigging conveyors in underground work, and mining methods are changing to allow the use of belts. Some steel mesh belts are in use because they are cheaper, but rubber belts are preferred; they are usually 26 or 32 inches wide, and have rubber covers on both sides so as to be reversible. Splices are generally made with steel wire hooks of the Adler or Nilos type; with a tool like a large pliers a set of hooks can be inserted in 3 to 8 minutes. Vulcanized splices are used, but require some hours. By means of change gears or variable-speed motors, conveyor speeds can be adjusted to the work; they are usually not over 300 feet per minute. Tandem drives on small pulleys are thought to hurt belt and splice; lagged snubbed pulleys are preferred. Lagging made of woven brake lining has been found satisfactory. It is clamped, not bolted, in place. Idler rolls of new type are sealed dust tight and water tight and hold some months' supply of grease. Idler stands are made of steel plate, no castings.*

* See *Glückauf,* March 24, 1934, and *Fördertechnik,* 1935, pages 235, and 1938, page 181.

CHAPTER 14

LIFE OF BELTS

The life of a belt depends upon many factors; a full discussion of them would be a review of what has already been said about belts and their accessory parts. The belt salesman is often called upon to answer the question: "How long will this belt last?" A prudent man will say that he cannot tell, that he is in the position of an actuary of a life-insurance company when asked how long a certain policyholder will live. He knows that some men start with a good constitution, wear out slowly, and live long; others are not so fortunate; they meet with accidents, neglect their health, and die young. So with belts; if the proper belt is selected for the work to be done in the first place, and if it receives proper care, it may be expected to have a long life, but if it is not suited to the work or if it is neglected or abused, it will not last long.

Many years ago when a rubber belt was merely "a rubber belt," men in the trade had the rule: A belt W inches wide with $\frac{1}{8}$-inch-thick cover loaded at one point only in a conveyor 100 feet centers should not wear out before it has carried 500 W^2 tons of coal. Nowadays, when makers and users recognize four or five grades of rubber belts (see page 55), such a statement is useless; any estimate of the tonnage life must start with the grade and amount of rubber in the belt, and then consider such factors as the length of the conveyor, the material handled and the size of the lumps, the hourly, daily, and yearly capacity of the conveyor, and the conditions under which it operates.

Table 14-A, based on the experience of a belt company long in the business, gives an approximate idea of the tonnage a rubber belt may be expected to carry before it is worn out, assuming average operating conditions and freedom from neglect, abuse, and serious accident.

Tonnage expectancy of a belt is naturally an uncertain subject, and the figures in Tables 14-A and 14-B are not to be considered as predictions; they can, however, be helpful as a guide in preliminary

studies and in determining whether a belt conveyor, or some other conveying means, should be used to solve a material-handling problem. Tonnage guarantees are not offered by belt manufacturers for the very good reason that they control only the making of the belt.

Why Belts Die Young. Companies in the business of making belts or selling belt conveyors are accustomed to receive complaints about the quality of belts somewhat in this style: "The belt we bought

TABLE 14-A

TONNAGE LIFE OF RUBBER BELTS IN THOUSANDS OF TONS

Width of Belt, inches	Conveyor Centers, feet									
	100	200	300	400	500	600	700	800	900	1,000
12	110	170	209	266	320	380				
14	160	240	297	378	459	540				
16	210	310	385	490	595	700				
18	260	390	484	616	748	880				
20	310	490	605	770	935	1,100				
24	430	720	910	1170	1430	1,690	1,950			
30	690	1130	1420	1820	2220	2,620	3,020	3,420		
36	980	1600	2000	2560	3100	3,670	4,230	4,780	5,340	
42	1340	2170	2730	3490	4250	5,010	5,770	6,530	7,300	
48	1740	2900	3680	4730	5780	6,830	7,880	8,930	9,980	11,000
54	2200	3630	4600	5900	7200	8,500	9,800	11,100	12,400	14,000
60	2750	4650	5940	7660	9380	11,100	12,820	14,540	16,270	18,000

Tonnage figures based upon:
Run of Mine Coal @ 50 lb per cu ft.
One loading point.
Top cover $\frac{1}{8}$-in. thick; 1400 to 2000 lb grade
Normal belt speed and rated capacity
Belt exposed to the weather.
Time limit 5 years.
Average operating conditions, freedom from neglect, abuse, and serious accident.
See Table 14-B for modifying factors for conditions other than those above.

from you on January 1st has lasted only six months and is going to pieces. Our previous belts all lasted two years; we think you should furnish a new belt or make some allowance on the price of the one we bought, etc., etc." Where it is possible to investigate such complaints, it will often be found that the cause is one of the items given in Table 14-C. For some of these, the seller of the belt may be responsible, but more often he is not. Some belt users present claims which are both absurd and unjust.

TABLE 14-B *

FACTORS TO BE APPLIED TO TONNAGE LIFE OF RUBBER BELTS

Belt Cover Quality	Per Cent	Materials	Per Cent
800–1000# grade	80	Coal, run-of-mine	100
1400–2000# do	100	crushed	120
2500–3000# do	120	pulverized	300
3500–4000# do	140	Coke, coke-oven size	50
		domestic sizes	80
Belt Cover Thickness	**Per Cent**	hot, coke wharf	10
		Stone and ore, 8 in. to fines	60
$\frac{1}{8}$ in. top cover	100	2 in. to fines	80
$\frac{3}{16}$ in. top cover	120	fines	90
$\frac{1}{4}$ in. top cover	140	Gravel, large	70
		small	80
Operating Conditions	**Per Cent**	Clinker, cool	70
		hot	20
		Slag	80
1 Feeding point	100	Chatt (zinc ore refuse)	50
2 Feeding points	80	Sand	90
Tripper	80	Cement, cool	110
Inclined	90	hot	40
Open conveyor	100	Grains	200
Housed conveyor	120	Salts	80
Single pulley head drive	100		
Tail or center drive	90		
Tandem drive	80		
Automatic weighted T. U.	120		

* Data in Tables 14-A and 14-B given to the authors by Mr William Warr of the Manhattan Rubber Company

TABLE 14-C

1. Buying the wrong grade of belt, one not suited to the work expected of it.
2. Neglecting to get the manufacturer's advice before buying a belt for difficult or unusual conditions.
3. Injury to belt by carelessness in getting it into place on the conveyor; cover or edges torn.
4. Splice not square with the belt; belt runs crooked, wears edges.
5. Material too hot; burns holes in the belt, chars the cotton.
6. Material corrodes cotton or softens rubber; plies come apart.
7. Loading chute of poor design; material wears the belt surface.
8. Skirt plates too long or badly set; belt cut.
9. Loading the belt directly over an idler; belt cut.
10. Speed too fast for proper pick-up of lumps; belt cut or abraded.
11. Speed so fast that belt carries only thin load; belt wears out in center.
12. Cover too thin for sharp lump material; belt cut, cover worn.
13. Handling wet, sharp material through a tripper; cover cut and worn.

14. Pulleys too small at drive, take-up, bend, snub, or tripper; plies come apart.

15. Spilled material allowed to accumulate around troughing idlers; rolls refuse to turn.

16. Using belt too thick or too stiff to lie down on the horizontal roll of the troughing idler; belt runs crooked, edges wear off.

17. Bad alignment of conveyor; edges of belt rubbed off, or worn against side-guide idlers.

18. Side-guide idlers badly set, or too many of them; wear off edges of belt.

19. Using fixed tripper where a plow would have been better.

20. Excessive take-up tension; belt stretches and splices pull out.

21. Using screw take-up where weighted take-up would have been better.

22. Exposure to sunlight; snow, ice, rain; rubber hardens, belt slips on driving pulley.

23. Frequent starting under full load; belt stretches and splices pull out. Slip injures belt surface.

24. Discharge chute choking; material catches on belt and tears it.

25. Decking not used or not replaced after having been removed to make repairs. A prolific cause of damage to belts.

26. If material carried is oily or if oil drops on the belt, ordinary rubber will soften and swell.

27. Pulley lagging rough or worn; bolts project, tear belt.

28. Using wrong belt to handle moist, soluble substances (sugar, salt, chemicals); liquid gets into fabric, crystallizes there, stiffens belt, plies come apart. Service requires balata belt or rubber belt with top, bottom, and edges well protected with covers.

The following actual complaints about belts may be of interest:

A belt at a by-product plant lost a strip of its cover; the complaint was based on "poor quality of rubber." Examination showed that the direction of tear was *with* the travel of the belt, so that it could not have been done by anything catching on the moving belt. Further investigation proved that the belt had been torn while it was being put on the idlers. On the resistance of rubber to tearing, see page 45.

A belt gave only 180 days of service instead of a year as other belts had done. It had been cut by a tool or something sharp falling on the belt.

A belt in a cannery ran in a tin-lined trough with plows at intervals to discharge the farmers' raw material into bins. The belt was injured by rough joints in the tin trough. When the material was dumped on the belt it contained nails, stones, and sharp sticks which the plows forced into the belt.

A belt handling raw fruit in a cannery. On account of bad alignment, the edges were worn and the acid juice corroded the cotton fibers.

A belt carrying wet sand. Edges rubbed off, water got in, plies came apart.

A belt carrying crushed ore. Ends not cut square at the splice; belt ran crooked, edges rubbed off, plies came apart.

A cement company ordered a belt without stating that it was for hot material. The rubber dried out and plies came apart. Belts can be specially built to handle hot products in cement mills.

A belt was used with a tripper to discharge wet gravel into a row of bins. The tripper had no brush; the lower pulley forced sharp particles into the belt, and the cover did not last long.

A coke belt lasted only one-third as long as was expected. The coke was not properly quenched; hot pieces burned holes in the belt.

A belt carrying crushed stone was reported to be wearing out rapidly. Investigation showed that it had been put on upside down with the ⅛-inch rubber cover in contact with the pulleys; the side that carried the stone had only the usual $\frac{1}{32}$ inch of rubber intended for the pulley side.

When a certain case of excessive wear on one edge of a belt was investigated it was found that the gauge of a tripper track was too great for the wheels, the tripper pulled out of square with the belt and the belt bore hard against the side-guide rollers in the tripper frame.

To handle wet coal under unfavorable conditions it had been found economical to use balata belts. One of these lasted only a few weeks and was replaced on the claim that it was defective. It was discovered later that hydrochloric acid spilled on a floor above the conveyor had dripped down on the belt.

A 30-inch belt carrying 6-inch stone lasted only one year. Investigation showed that head pulley was not crowned and user had placed side-guide idlers every 10 feet. When these errors were corrected, belts lasted three years.

An 18-inch belt carried foundry sand up an incline, then on a level. After six weeks it started to crack. The bend at the hump had been made over *one* troughing idler, and the crack corresponded to the angle between two idler pulleys. The idler was replaced by a 20-inch face pulley and the trouble stopped.

A 24-inch belt, taking stone from the head of an elevator, lasted only six weeks. The feed was at right angles to the belt. When the foot of the belt was lowered and feed delivered *with* the run of the belt, belts gave good service.

A 30-inch belt on 20° incline would not run straight. The head pulley had worn so much that the face was hollow instead of convex. A band was fastened on, turned to a crown, then the belt ran straight.

A belt handling coal lasted only five months; others had lasted three years. Investigation showed idler pulleys stuck tight; belt sliding over them had worn the rims entirely away.

A belt used in a magnetic separator had a short life. A short circuit in the windings in the head pulley had heated the pulley and cooked the life out of the belt.

A user complained that his 36-inch belt, carrying 3-inch stone, had "rotten duck." A section of decking had not been replaced after being removed to permit the making of repairs, with the result that stone fell through, jammed under foot pulley, and punched holes in belt.

A belt carrying sintered ore showed ply separation for its whole length in a width of 2 inches. A sharp fin of sinter had built up on one return idler, and the constant folding back and forth had destroyed the friction of the belt.

A friction-surface belt was used to carry sugar, and soon cracked. The

service required a balata belt, or a good rubber-covered belt. (See Table 14-C, item 28.)

A belt spilled much material and had one edge worn off. Conveyor was 6 inches out of line, one edge of belt riding off the idlers, the other down in the trough and spilling. When conveyor was re-aligned, trouble stopped.

Cotton seed spilled from a belt piled up under a floor and buried the return run. When the plant was idle, the seed rotted, and products of fermentation destroyed the belt.

A user ordered a 9-ply belt, then complained that it ran crooked. Investigation showed that his idlers had a deep trough and belt would not lie down on horizontal rolls. If the user had stated his case, belt makers would have advised a 7-ply belt as sufficient for the work and better suited to the idlers.

Comparison of Life of Belts. The true measure of belt service is what the belt will do per dollar of investment or, in other words, what it costs per ton of material carried, or per year or month of service rendered. There is no standard of comparison for this because operating conditions in any two plants are never exactly alike; what may be extravagance in one plant may be justifiable practice in another. A certain 36-inch 8-ply rubber belt in a western smelter carried over 7,000,000 tons of ore in thirty-eight months at a cost of less than 0.02 cent per ton—a most excellent record. On the other hand, if a belt used in the throwing device shown in Figs. 9-24 and 13-3 lasts one week and carries 30,000 tons at a cost of 0.4 cent per ton, it is considered excellent service, although the rate per ton is very much higher.

It is often unfair to judge the quality or the suitability of various kinds of belts, rubber of different makes, stitched canvas, balata, or solid-woven belts by a comparison of the life of one or two specimens. As has been stated, many belts must be replaced because of abuse or accident, not on account of wear in service. No belt, whatever the type or make, is proof against all the items mentioned, and most of the mishaps, if they occur, will shorten the life of any belt.

Stitched Canvas Belts in Service. Manufacturers of stitched canvas belts have furnished the following information:

At a sand and gravel washery plant in northern Illinois 300 feet of 30-inch 8-ply stitched canvas belt ran four seasons of eight months each and handled 1,497,000 tons of sand and gravel—a record for the plant.

At another plant of the same kind 400 feet of 24-inch 8-ply stitched canvas belt on an incline lasted six seasons of eight months each and handled 1,235,000 tons of sand and gravel. The long life of this belt may be attributed in part to the fact that it was not troughed but ran flat.

A stitched canvas belt, 18-inch 6-ply, 254 feet long, saturated with a class 2 compound (see page 62), lasted one year in an Illinois cement plant and

carried 143,000 tons of hot cement, which was 4000 tons more than the combined tonnage of two belts previously used. The canvas belt was replaced by another kind of belt under a guarantee of equal service, but the makers of it had to furnish four belts during the year to make good the guarantee. When the year was up, the cement company put in another canvas belt.

A 36-inch 6-ply canvas belt on an inclined conveyor, 430-foot centers, rising 80 feet, speed 450 feet per minute, over 22½° troughing idlers with snub drive from 54-inch head pulley, carried 1,400,000 tons of run-of-mine coal at a cost of less than ⅕ cent per ton.

A 36-inch 8-ply canvas belt on 22° incline, 270-foot centers, running over 20° troughing idlers, 54-inch lagged head pulley, carried 1,000,000 tons limestone crushed to 6 inches and less.

A 24-inch 5-ply canvas belt, horizontal, carried 723,000 tons wet screened and washed coal for less than ⅒ cent per ton.

A 30-inch 8-ply oil-treated belt on a 240-foot centers conveyor, inclined 15° and driven by tandem pulleys at the foot, carried 850,000 tons of blast-furnace slag in pieces up to 200 pounds weight. This was considered a good record for such service.

A 22-inch 6-ply asphalt-treated belt, 163 feet long, carried 400,000 barrels of hot cement clinker, temperature from 200° to 300° (about 45,000 tons). This is considered a good record.

CHAPTER 15

WHEN TO USE BELT CONVEYORS

General Advantages of Belt Conveyors. The range of uses of belt conveyors is very wide; they carry all kinds of bulk materials from clippings of tissue paper to ore in pieces weighing 200 to 300 pounds. Package goods of all sorts except the heaviest bales and barrels are successfully handled.

For many of these uses, the belt conveyor is the cheapest and best machine; for some it is the only machine. It may well be the only machine that will give the required capacity. Capacity is generally a matter of speed; in a belt conveyor, it is comparatively easy to get high capacity by using a speed high enough, for, so far as the driving of the belt is concerned, the possible speeds are far beyond all needs for conveying. Belts used in the transmission of power often travel 5000 feet per minute, but few belts except some used in grain conveying travel more than 600 feet per minute. Chain conveyors, on the other hand, seldom travel over 250 feet per minute; at higher speeds, the shock and noise caused by chain links engaging with the sprocket wheels become objectionable and the wear in the chain joints becomes troublesome.

The machinery of a belt conveyor is usually simple and light in weight; it is not likely to break down without warning, and it generally consumes less power for the work accomplished than any other form of conveyor.

The characteristics mentioned above have been responsible for the great development and wide use of belt conveyors. In the choice of a conveyor to do a certain work in a particular place they should be carefully considered, but at the same time some collateral disadvantages should not be overlooked. These do not apply to belt conveyors in general, but rather to specific uses as referred to below.

When to Use Belt Conveyors. Since a belt conveyor will carry material horizontally or up an incline, or even do both in one machine, it will be of interest to discuss the merits of belt conveyors as compared with those of conveyors of other types and with combinations of elevators and conveyors.

Boiler Houses. Distribution of coal in overhead bins can be accomplished by belt conveyor or flight conveyor. If the bin is a long continuous one, a belt with a traveling tripper will fill it, but if the bins are short, with spaces between, or separate round tanks, a flight conveyor is more convenient for the discharge. Separate fixed trippers could be used, but they wear out the belt by repeated reverse bending and by throwing material back on the belt when coal is to be carried past one or more of them. To move a traveling tripper over clearance spaces while the belt is empty and set it over various bins requires care and attention on the part of the attendant and may cause delays in the operation of the machine. Traveling trippers with large storage chutes to carry the discharge from the belt past clearance spaces and then drop it into the separated bins have been designed and patented, but are not in commercial use.

If the capacity required is less than 50 tons an hour, the work is less than what a 14-inch belt will do at a moderate speed. Twelve-inch belts are practically obsolete; 14-inch belts are hard to load with coal larger than 2-inch size. For small capacities a belt conveyor will cost more per foot run and will be burdened with the expense of the tripper.

If the capacity is 100 tons an hour or more, belts 20 inches or more in width can be used. These are easier to load with crushed coal, and they run straighter than narrow belts. Belt conveyors in this class, as a rule, cost less than good flight conveyors and they consume less power.

If the conveyor is shorter than 75 or 100 feet a flight conveyor may be better. Its entire length can be used for distribution; but in a belt conveyor, 15 or 20 feet is lost between the loading point and the point of first discharge to prevent the tripper from lifting the belt under the chute (see page 236). A short belt wears out faster than a long belt, and since the terminals of a short conveyor use more power than the run of the belt, a belt conveyor in this class may not show any measurable saving of power as compared with a flight conveyor.

If the length of the conveyor is more than 200 feet, a belt will show a noticeable saving of power, and for lengths greater than 300 feet the expense of a belt conveyor is likely to be less than the cost of a double-strand conveyor with roller chains. A flight conveyor on a single chain, however, will cost somewhat less than the belt, but it will not handle large coal so well. It will require about twice as much power as a belt conveyor.

If quietness of operation is essential, use a belt. It must be said, however, that modern roller flight conveyors and double-strand roller-chain conveyors are much quieter than old-style flight conveyors.

If the conveyor is close up under the roof and receives from an elevator, a flight conveyor may fit in better. It requires less headroom than a belt with a tripper; the loading chute can be shorter and can load the conveyor from the side. A belt should not be loaded from a side-delivery chute; to load it properly requires more height than to load a flight conveyor.

The belt is the costly item in a belt conveyor and at the same time the most vulnerable part. If it receives good care it will, in places suited to it, render service at a lower cost per ton of material carried than any ordinary flight conveyor. But the belt may be damaged or ruined by a number of causes, and then the charges for repair or renewal may be excessive. The ordinary causes of belt failure are given in Table 14-C.

A corresponding list of causes of failure of flight conveyors would be shorter. In general, a flight conveyor will stand more abuse and will work under bad conditions and under a lack of care that would be harmful to belts.

Transfer of Material without Distribution by Tripper. In this case the belt makes a better showing than a flight conveyor except for short distances and low capacities. For feeders, a corrugated steel apron is more rugged than a belt. It is less likely to be injured by tools, sticks, and hard, sharp lumps.

The belt, however, will make a cleaner delivery of material at the head end and will not spill so much on the return run.

On inclines, a flight conveyor will work at angles steeper than 25°, but a belt will not. For heavy tonnages a steep flight conveyor becomes costly and, as between the two, it may be better to use a longer belt at a flatter angle, or even two belts in series to raise the material to the required height. Transfer of coal at by-product plants is done chiefly by belt. Here the distances are great and the capacities high. Flight conveyors would cost more to build and be more costly to drive. The disadvantages of belt conveyors in this work are the high first cost of the enclosing structures and their maintenance and the fact that the bunkers and the loading points are spread apart over a stretch of ground which would otherwise be unnecessary.

Crushed Ores, Gritty Materials. Belts are in general use for this work.

Lump Rock, Sand, Gravel, Excavated Earth. Belts do this work better than any metal conveyors, except that, for very large rock, wide apron conveyors with wood or steel slats will stand abuse that would ruin belts. They cannot, however, be economically made as long as belt conveyors.

Ashes. Many belts tried on this work have failed from damage by hot cinders. Moreover, if the ashes are handled wet, the cinders cut the cover and the water ruins the duck.

Coke. Belts are in general use. The grit wears out metal conveyors too fast for economy. Coke is too friable to handle in scraper conveyors.

Wood Chips. Belts are in general use. The work is light and does not require expensive belts.

Package Handling. Belts are in general use for all except the largest packages and the roughest work.

Zigzag Conveyors for Elevating Material. In a few places a rising series of short inclined belts have been used to do the work for which a bucket elevator or a skip hoist would ordinarily be installed. They do not save in first cost; they take up more space and require more power than a single elevator. Some have been satisfactory to owners; in others the short belts wore out rapidly from slip due to the steep incline and from abrasion at the loading point.

In the matter of spare parts to be kept on hand, there is this difference. Repair parts for a flight conveyor can be kept on hand indefinitely, but a spare rubber belt should not be ordered too long before it is really needed, or it will deteriorate in storage. Canvas belts suffer less in this respect.

Comparisons. Where comparisons have been made between belt conveyors and chain conveyors on the basis of tonnage, length, or cost, the limits are not hard and fast, and between them there are chances to exercise some personal preference. Both kinds of machines have been on the market for many years and have been to a great extent standardized. There is no "universal conveyor"; for handling heavy or gritty ores as in the western smelters or in conveying heavy tonnages of coal over long distances, a flight conveyor is not generally considered, but in many places, boiler houses especially, it makes the best distributor even though it does take more power to run it. The older forms of flight conveyor were noisy, but modern roller flight conveyors, or roller chain conveyors, are, by comparison, nearly noiseless; they are strong and rugged, easy to load, easy to discharge, spill less dirt outside, and they will work under bad conditions where a belt will fail. Some engineers have a prejudice against scraping coal in a trough. It does take more power, but it does not hurt the coal, even anthracite coal; and as for the wear on the trough the item of replacement is not an important one.

The right choice of a conveying machine depends upon:

1. A proper knowledge of the limits within which each kind works

best. This implies some acquaintance with the disadvantages of each type of machine. The advantages are usually stated in manufacturers' catalogs, proposals, and advertising, but the disadvantages are generally learned by experience.

2. A recognition of the fact that a machine highly successful in one place may be a failure in another place, even at the same kind of work.

3. Proper consideration of the auxiliaries necessary to the operation of the conveyor. A simple conveying medium with complicated auxiliaries for feeding or discharge may be less desirable than a more complex conveying medium with simpler auxiliaries. In a belt conveyor, for instance, the belt itself is simple and strong, but it requires the right kind of a chute to load it safely; and to effect a discharge at intermediate points, a tripper, with its many parts, may be necessary. A flight conveyor has a chain made up of many parts, but feeding is simple and discharge is merely dropping the material through a hole in the trough. A screw conveyor is not economical of power, but for low cost, simplicity of mounting, compactness, ease of feed and discharge, and cleanliness and avoidance of dust, it is in many places preferable to any other kind of conveyor.

Long-distance Conveying by Belts. On this subject, see pages 184 and 270.

SECTION II.—BELT ELEVATORS

CHAPTER 16

GENERAL DESCRIPTIONS

The Elements of a Belt Elevator. A belt elevator for bulk materials consists of:

1. Buckets to contain the material.
2. A belt to carry the buckets and transmit the pull.
3. Means to drive the belt.
4. Accessories for loading the buckets or picking up the material, for receiving the discharged material, for maintaining belt tension, and for enclosing and protecting the elevator.

Kinds of Belt Elevators. Any kind of belt with buckets attached can be run around an upper pulley and a lower pulley, and it will elevate loose material. If the belt speed is high enough the contents of the buckets will be thrown out in passing over the upper pulley (head pulley) and will fall into a chute set to clear the descending buckets, some distance below the head shaft. This is a centrifugal discharge elevator (Fig. 16-1); it may be vertical or it may stand at an angle. Vertical elevators depend entirely on the action of centrifugal force to get the material into the discharge chute and must be run at speeds relatively high. Inclined elevators with buckets spaced apart or set close together may have the discharge chute set partly under the head pulley, and since they do not depend entirely on centrifugal force to put the material into the chute, the speeds may be relatively lower.

Nearly all centrifugal discharge elevators have spaced buckets with rounded bottoms; they pick up their load from a boot, a pit, or a pile of material at the foot pulley.

If the buckets are triangular in cross section and are set close on the belt with little or no clearance between them, the machine is a continuous bucket elevator (Fig. 16-2). It can be run at high speed with centrifugal discharge as in some grain elevators, but this is not common. The chief use of continuous bucket elevators is to carry difficult materials at slow speed. Discharge is aided slightly by centrifugal

force, the contents of each bucket pouring out over the inverted bottom of the bucket ahead of it, and into the head chute. The elevator may be vertical or inclined; to permit the buckets to be loaded directly from a chute, most elevators of this kind are inclined; very few pick up their load under the foot wheel.

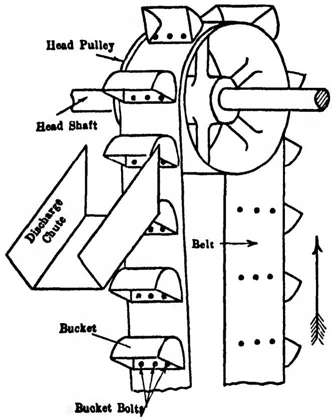


FIG. 16–1. Head of Centrifugal Discharge Belt Elevator.

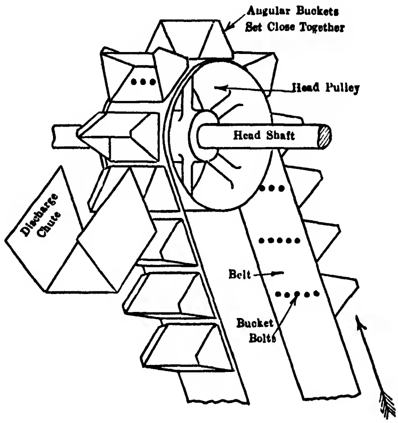


FIG. 16–2. Head of Belt Elevator with Continuous Buckets.

Elevator Buckets. The general requirements for an elevator bucket are as follows:

1. Dimensions large enough to pick up, hold, and discharge the largest pieces of the material handled by the elevator.
2. Cubic contents enough to give the required elevator capacity in pounds per minute, or tons per hour or per day, considering the speed of the belt, the bucket spacing, the regularity or irregularity of loading, and the probably incomplete filling of the buckets.
3. Strength and stiffness to pick up its load without crushing or distortion.
4. Thickness of metal sufficient to resist wear to an economical degree.
5. Inside of bucket so shaped that material will not stick there and fail to discharge.

To meet these requirements and to handle the many kinds of bulk materials, buckets for belt elevators are made in several styles:

1. Buckets with rounded bottoms, common in centrifugal discharge elevators.
2. Buckets with angular bottoms, sometimes used in high-speed

centrifugal discharge elevators for grain, but more often employed in continuous bucket elevators for coarse, heavy materials.

With respect to their construction and the material of which they are made, elevator buckets may be classified thus:

1. One-piece or three-piece buckets of tin plate or light-gauge sheet steel, with seamed corners and reinforced by steel bands. These are used for flour-mill products and for grain.

2. One-piece buckets of heavier steel, pressed to form and riveted or welded, without reinforcing bands. Used for grain and for materials heavier than grain.

3. Hot-pressed or cold-pressed seamless sheet-steel buckets, used for grain and other bulk materials not too heavy.

4. Cast malleable-iron buckets, for coal, ores, minerals, and other coarse, heavy materials.

5. Two-piece or three-piece buckets of heavy steel plate made with angular bottoms for continuous bucket elevators.

Elevator buckets are described more fully in Chapter 18.

Elevator Belts. The general requirements for an elevator belt are:

1. Sufficient flexibility to wrap easily around the head and foot pulleys.

2. Width enough to fasten the elevator buckets securely and to avoid twisting or turning over on the ascent.

3. Thickness sufficient to transmit the working pull without excessive stretch, to back up the buckets without deflection, and to resist the tendency of the bucket bolts to pull through the belt.

4. A protective cover or a body of fabric thick enough and strong enough to resist, to an economical degree, the surface wear in elevators that handle sharp, abrasive materials.

Practically all elevator belts in this country are made of cotton fiber in some form; they are:

1. Rubber belts.
2. Stitched canvas belts.
3. Balata belts.
4. Solid-woven belts.

These are described briefly in Chapter 1 and more fully in Chapters 3 and 20. Leather makes a good elevator belt for dry work; it was in general use for that purpose up to about 1870, but now fabric belts are cheaper, more economical, and better suited to most conditions. Elevators with woven-mesh steel-wire belts have been used in Europe for light service, but they are unknown in this country. Ele-

vators with buckets fastened to two or more parallel strands of wire rope have been tried at various times but without success.

Driving the Belt. The elevator belt is driven by the frictional contact between it and the rim of the head pulley, and since it is not possible to use a binder pulley or snub pulley against the bucket side of the belt, the angle of contact is limited to about 180°. The ability of the head pulley to drive the belt depends (see page 150) on the angle of belt wrap and the coefficient of friction between the belt and the pulley rim; in an elevator both of these are fixed within certain limits, and it is not so easy to increase the driving effect as in a belt conveyor (see page 151) where the angle of wrap can be made larger than 180°. To get more pull in an elevator it is necessary to put tension on the belt, relatively more than in conveyors, and this leads to the use of higher unit stresses in elevator belts than in conveyor belts.

Belt elevators have been driven at the foot, but the drive is always uncertain and often troublesome (see page 378).

Accessories for Loading the Buckets. In some forms of elevating and conveying apparatus it is possible to deposit separate charges or loads in consecutive buckets as they pass, by means of a mechanical loader, but at the speeds at which centrifugal discharge elevators run that cannot be done; the material cannot be guided into a bucket moving at a rate of 3 to 10 feet per second, and the impact would scatter and spill it. At the lower speeds of continuous bucket elevators, 80 to 200 feet per minute, the difficulties of mechanical loading are less, but still serious enough to make this process expensive and troublesome. It is much easier to load continuous buckets by means of a chute, especially when the elevator is inclined, so that what one bucket misses, the next one will catch. In centrifugal discharge elevators, however, it is never possible to load buckets from a chute without spill, and it is not often attempted; it is necessary in all cases to let the buckets pick up some or all of their load as they pass around the foot wheel or as they enter the vertical run. If the elevator digs from a pit or a pile, the material is naturally confined to the path through which the buckets sweep, but otherwise a box or boot is used to form a mounting for the foot shaft and keep the material within reach of the buckets.

Belt Tension. Usually the foot-shaft bearings are adjustable in position either as take-up bearings separate from the boot or as sliding bearings which form part of the boot. Sometimes the foot-shaft bearings are fixed; then the take-up bearings are placed at the head of the elevator.

Discharge at the Head. In some forms of chain elevators the buckets discharge on the lift, but all belt elevators discharge at the head into a chute set to catch the material, either as it is thrown out by a centrifugal discharge elevator (Fig. 16-1) or as poured out by a continuous bucket elevator (Fig. 16-2). The position of the chute and the discharge of material from the buckets depend upon three factors:

1. The speed of the belt.
2. The diameter of the head pulley.
3. The spacing and shape of the buckets.

At the same time, the loading or pick-up at the foot depends upon:

1. The speed of the belt.
2. The diameter of the foot pulley.
3. The spacing and shape of the buckets.

The best speeds for the pick-up and discharge of different materials have been determined by trial and experiment, and have been confirmed by years of successful practice. They agree so well with results given by analysis that it will be of interest to show how they can be established by some consideration of the theory of the subject. This discussion will at the same time serve as an introduction to the further consideration of the design and construction of belt elevators.

CHAPTER 17

CENTRIFUGAL DISCHARGE ELEVATORS

Pick-up and Discharge of Elevator Buckets. When a mass of material of weight W is passing around a wheel it is under the influence of two forces: one, gravity, acts vertically downward with a force W; the other, centrifugal force, acts radially outward from the center of rotation with a force $= \frac{Wv^2}{gR}$, where v = velocity of the mass in feet per second, g = acceleration of gravity = 32.2 feet per second, and R is the radius in feet to the center of rotation.

The action of buckets passing under a foot wheel or over a head wheel is shown in Fig. 17-1. In position 3, centrifugal force acts horizontally outward, gravity acts downward, and the diagonal resultant obtained by completing the parallelogram of forces shows by its *direction* that the resultant pressure is downward within the bucket, and, by its *length* on the scale to which the other forces are drawn, that the pressure is $\sqrt{2} = 1.414$ times the weight of the mass in the bucket if centrifugal force and weight are equal in amount. At 4, the pressure decreases; at 5 it becomes zero if the two forces are equal; and at 6 it acts to propel the mass from the bucket toward a chute set to catch the material and with a force which on the scale of the drawing equals about three-fourths of the force of gravity.

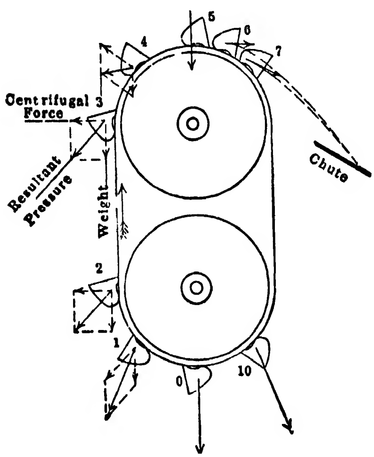


Fig. 17-1. Action of Forces on Pick-up and Discharge. Centrifugal Discharge Elevators at Speeds Given in Table 17-A.

Best Speed for Elevator Discharge. In order to deliver material to the chute without spilling or scattering, it would seem that, at a place near the top of the wheel the two forces, weight and centrifugal force, should be equal in amount, because then, for position 5, the mass within

the bucket will be in equilibrium or a state of suspension, neither tending to fly out upward nor to fall out on the wheel, but ready to move out freely when the resultant of the two forces on the descending side of the wheel urges the material toward the mouth of the bucket. That condition of equilibrium exists when

$$W = \frac{Wv^2}{gR}, \text{ (see above) or } v^2 = gR \tag{1}$$

or since

$$v = \frac{2\pi RN}{60}$$

where N = number of revolutions per minute, then

$$N = 54.19\frac{1}{\sqrt{R}} \tag{2}$$

This relation between radius of the head wheel and its revolutions per minute holds good in practice for the discharge of liquids or dry free-flowing materials like grain from elevator buckets of ordinary shape.

Table 17-A, calculated from equation 2, shows diameters of head wheels and corresponding speeds for the entire range of sizes used in

TABLE 17-A

HEAD WHEELS AND SPEEDS, CENTRIFUGAL DISCHARGE AT HIGH SPEED

Diameter of Wheel, inches	Revolutions per Minute	Belt Speed, feet per minute	Diameter of Wheel, inches	Revolutions per Minute	Belt Speed, feet per minute
12	69	217	42	39	429
15	62	247	48	37	465
18	56	264	54	35	495
21	53	292	60	33	518
24	50	314	66	32	553
27	47	333	72	31	584
30	45	353	84	29	637
33	43	372	96	27	679
36	41	386	108	25	707
39	40	408	120	24	754

centrifugal discharge elevators. In the calculations, assumptions have been made for the thickness of belt and the projection of buckets ordinarily used with each size of head wheel, and R has been taken as

the distance from the center of the head shaft to the center of gravity of an average load in the bucket.

In this discussion, a centrifugal discharge elevator is one that runs at a speed high enough to discharge the contents of the buckets clear of a vertical run of descending buckets.

The parabolas drawn from positions 6 and 7 (Fig. 17-1) represent the path of discharge from buckets in those positions. The material all clears the head wheel and will enter a chute with its upper end placed to clear the descending buckets and about 10° of arc below the center of the wheel. The figure shows successive positions of one bucket, not simultaneous positions of adjacent buckets.

The upper half of Fig. 17-1 represents the action on a head wheel, and the lower half shows what happens on the bottom of a foot wheel. If the material were fed in on the descending side of the foot wheel, a bucket at position 10 could not retain it because the resultant is directed toward the mouth of the bucket. At position 0 the resultant falls within the bucket, but nearly all the material would flow out because its surface would tend to lie at right angles to the line of pressure, just as in a vessel of water whirled around an axis, the water surface tends to stand at right angles to the line of pressure exerted along the radius of rotation. At 1 the tendency to squeeze material out of the bucket is less than at 0, and at 2 it is still less; but even there it would be impossible for a bucket to retain a full load under the influence of a pressure which is 1.4 times the force of gravity and is directed at such an angle that when the material did shift to square itself with the line of pressure some of it would fly out over the front lip of the bucket.

These considerations show that it is impossible for buckets to carry free-flowing material like grain around or from under a foot wheel when the speed of travel is that given in Table 17-A. But if the boot is so arranged that the material is fed in above the level of the foot shaft, as in Fig. 17-6, then the bucket is loaded on a straight lift above position 2, where centrifugal force no longer acts on it. When a boot is fed at the back, the material is swept around by the buckets, but the buckets fill at the front (see Fig. 17-8).

Effect of Higher Speeds. Fig. 17-2 shows the effect of making centrifugal force equal to twice the force of gravity, a condition which exists when for a wheel of given size the revolutions per minute $= 1.414 = \sqrt{2}$ times the values given in Table 17-A. Considering first the upper half of the diagram, as representing the top of the head wheel, we see that the contents of a bucket rising to the position 3 will be acted upon by a force which is $\sqrt{5} = 2.23$ times as great as the

force of gravity. The direction of the force shows that some of the material must be suddenly spilled over the front lip of the bucket. Once over the lip, the spill will fly outward and fall down the rising leg of the elevator. The resultant pressure decreases at 4, but at 5 the direction of the pressure is vertically upward and some of the contents of the bucket would be thrown in that direction. If the scale of the drawing represents a 96-inch wheel making 38 revolutions, the spill would rise about 3 feet and then fall down. Any material remaining in the buckets at 6, 7, 8, and 9 would pass off in the parabolic curves shown in the figure.

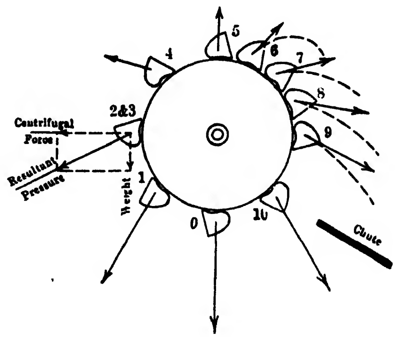


FIG. 17-2. Action of Forces on Pick-up and Discharge. Centrifugal Discharge Elevators at Speeds 41 Per Cent Higher than Table 17-A.

The discharge of some materials at very high speed is affected by the air current set up by the pulley and the buckets. This "fan action" limits the speed at which dusty materials can be handled (see page 305). For its influence on grain elevators, see page 311.

Since the direction of the resultant of pressure shows the direction in which the material starts to leave the bucket, and since for positions 5, 6, 7, and 8 (Fig. 17-2) these lines do not point directly toward the mouth of the bucket, there is a possibility that some of the material will be trapped within the bucket and descend with it, or at least require the chute to be set low to catch the tail end of the scattering discharge.

When the lower half of Fig. 17-2 is compared with the lower part of Fig. 17-1 it appears that the pressures which would force material out of the buckets under the foot wheel are greater in amount and even less favorable in direction at the higher speeds. The lower half of Fig. 17-2 would represent the action of forces in an elevator boot where the foot pulley is half as large as the head wheel, but if the pulley is only one-third or one-fourth as large as is usual in large grain elevators the centrifugal forces are then three or four times as great at the foot as at the head since the force varies as $\frac{1}{R}$ for a constant value of v (belt speed). Such conditions are even more unfavorable for filling the buckets unless the loading is done after the buckets are on the straight vertical lift, as in Fig. 17-6.

Fig. 17-3 shows, on a smaller scale, a grain elevator with a 96-inch head wheel and a 24-inch foot wheel. The resultant pressures (shown to proper scale) which prevent the grain from entering the buckets while they are in contact with the foot wheel are six or seven times as great as those which throw the grain out of the buckets on the head wheel. It is quite evident that the buckets can take no load below the

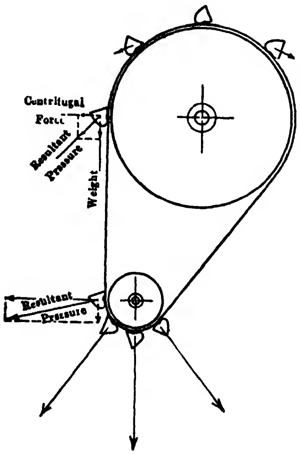


FIG. 17-3. Pressures Affecting Pick-up in a Grain Elevator with Small Foot Pulley

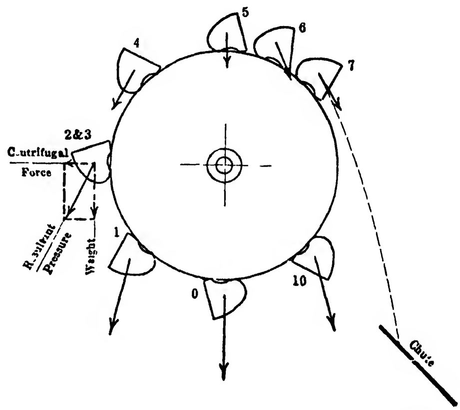


FIG. 17-4. Pick-up and Discharge at Speeds 30 Per Cent Lower than Table 17-A.

level of the foot shaft, but must be filled after they are on the vertical run.

Effect of Lower Speeds. Fig. 17-4 shows the effect on pick-up and discharge when the speed is lower than the best speed in the ratio $1 \text{ to } \frac{1}{\sqrt{2}} = 1 \text{ to } 0.7$, a condition which makes centrifugal force one-half the force of gravity and N about seven-tenths the values given in Table 17-A. Considering the upper half as representing the top of the head wheel, a loaded bucket rising to the position 3 will not spill any of its load because the resultant pressure differs very little from W (due to the force of gravity) either in amount or in direction. The same condition exists at 4, but at 5 there is a downward force equal to $\frac{W}{2}$ which would tend to spill some material out of the bucket onto the top of the wheel. At 6 the resultant shows that more would spill out, and at 7 the remaining material would be thrown clear of the wheel and could be caught in a chute, the upper end of which is set to clear

the descending buckets and on about 45° of arc below the center of the wheel.

Such a discharge can be used for an inclined elevator, but it is too slow for a vertical elevator.

For a discussion of the discharge from inclined elevators, and of the point at which discharge begins see Chapter 24 and Fig. 24-7.

Belt Speeds for Different Materials. From what has been said it is evident that the figures of Table 17-A apply to free-flowing materials which can be piled deep in a boot and to buckets which finish their loading while on the vertical run above the center of the foot wheel.

It does not apply to substances like coal, ores, minerals, and cinders which are not free-flowing, or which contain lumps, or which are likely to be wet and stick to the buckets. It is not safe to pile such materials deep in a boot; the work of pulling buckets through the mass would be wasteful of power, and buckets would be broken or torn loose by the severe strain. When the materials of that kind are handled in a centrifugal discharge elevator with spaced buckets, the feed must be tangent to the sweep of the buckets and not so high up that the foot wheel would be dangerously buried if the amount of feed should exceed the lifting capacity of the buckets for a short time. It is customary to use sloped front boots (Fig. 23-9) for such work, and the travel of the buckets *while on the foot wheel* must be limited to a speed at which centrifugal force will not throw material out of the bucket at positions 1 and 2 (Fig. 17-1). This tendency to throw material out is resisted at position 1 by the pressure between the moving bucket and the material flowing into the boot or lying on the bottom of the boot; and at position 2 by the fact that coarse substances like coal and minerals are not so free-flowing as to be squeezed out of the bucket by a resultant pressure of the direction and intensity shown in Fig. 17-1.

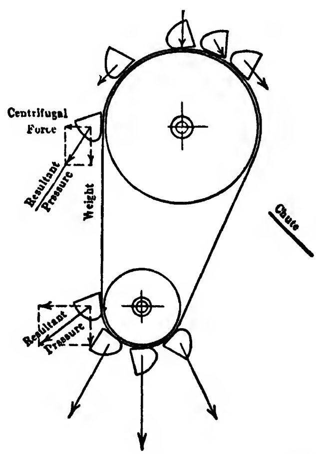


FIG. 17-5. Pick-up and Discharge at Speeds Given in Table 17-B. Foot Wheel Half as Large as Head Wheel.

Nevertheless, it is not practicable to use the speeds of Table 17-A for coal, ores, minerals, cinders, and similar coarse materials, and there are other reasons why that table should not be used for dusty materials. For these various materials, and even for free-flowing materials under certain conditions, it

is proper to use the speeds of Table 17-B which are 82 per cent of those of Table 17-A. At these speeds centrifugal force is two-thirds of the force of gravity, and the conditions of discharge are shown in the upper part of Fig. 17-5.

Reasons for Table 17-B. Some important reasons are given in a paragraph above; to these may be added the following:

1. For various reasons, foot wheels are often made smaller than head wheels. Centrifugal force is then greater than at the head, since for the same belt speed or velocity v, centrifugal force varies as $\frac{1}{R}$ (see page 297); that is, it is twice as great for a wheel half as large. That difference, in the greatest degree existing in practice, is shown in Fig. 17-3, but even where the belt speed is less and the difference between the foot wheel and the head wheel is not so great, the action at the foot may be such as to prevent buckets from loading properly. Fig. 17-5 shows to scale what happens in an elevator for coal, etc., where the speed is according to Table 17-B and the foot wheel half as large as the head, say, 18 and 36 inches, respectively. The pressures due to centrifugal force which hinder the filling of the buckets while they are on the 18-inch foot wheel are more than three times as great as those which throw the material out of the buckets while they are on the 36-inch head wheel. If the foot wheel were 24 inches in diameter (ratio ⅔) the pressures would be the same in intensity and direction as those shown in Fig. 17-1; if the wheel were 27 inches (ratio ¾) they would be more favorable for the filling of the buckets; if the

TABLE 17-B

HEAD WHEELS AND SPEEDS, CENTRIFUGAL DISCHARGE AT MODERATE SPEED

Diameter of Wheel, inches	Revolutions per Minute	Belt Speed, feet per minute	Diameter of Wheel, inches	Revolutions per Minute	Belt Speed, feet per minute
12	56	176	39	33	337
15	51	200	42	32	352
18	46	217	48	30	377
21	43	237	54	28	396
24	41	257	60	27	424
27	39	276	66	26	449
30	37	290	72	25	471
33	36	311	84	23	506
36	34	320	96	22	554

wheels were of the same size, i.e., 36 inches, the opposing pressures would be still less, the buckets would take their load lower down in the boot, with less turmoil and stir in the material, and they would carry, in general, a fuller load.

2. If the materials are picked up and discharged at high speed, the wear on buckets is serious; there is the risk of damaging them or tearing them loose, the strain in the elevator belt or chain is likely to be injurious, the wear on the head chute may be objectionable, and with

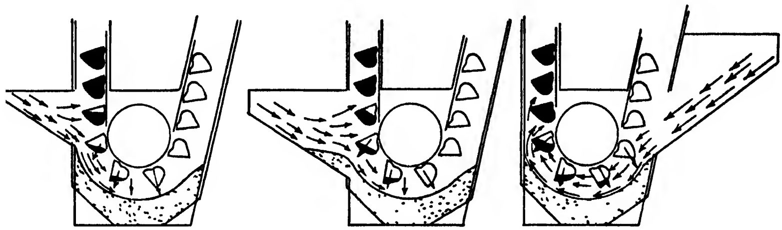


FIG. 17-6. FIG. 17-7. FIG. 17-8.

FIG. 17-6. Pick-up in High-speed Grain Elevator, with Front Feed at Level of Foot Shaft.
FIG. 17-7. Front Feed below Level of Foot Shaft.
FIG. 17-8. Back Feed.

some friable materials, like coal, the breakage from striking the chute violently may be a disadvantage.

3. Overcoming the friction losses at high speeds means a waste of power.

Practical considerations like these have established the following rules:

High-speed Grain Elevators. If the material is dry, granular, free-flowing, not abrasive, not damaged by picking up or throwing down at high speeds, and if high capacity is desired, use Table 17-A.

This is pre-eminently the table for high-speed, high-capacity grain elevators.

Boots for high-speed grain elevators should have the front feed inlet so high and the front of the boot and the lower part of the casing so shaped that the buckets can complete their loading after passing above the center of the foot wheel in its upper position. Fig. 17-6 shows how grain is picked up in such a boot; if the feed in front is low, as in Fig. 17-7, the pick-up is not improved, the buckets may not take a full load, some of the grain may lie comparatively dead in the feed hopper, or the chute may be choked. If the feed is at the back (Fig. 17-8) there is no gain in placing the chute high, but if it is tangent

to the sweep of the buckets in the upper position of the foot wheel, then, if the elevator is choked to a standstill before the feed is stopped, less grain will flow into the boot and less will have to be shoveled out to clear the boot in starting up again.

Discharge chutes for high-speed grain elevators may be set so that the lip just clears the buckets and lies on an arc 15° or 20° below the center of the head wheel. On this point, see page 401.

Limitations of High Speed. So far as discharge is concerned, Table 17-A applies to all kinds of centrifugal discharge grain elevators, but there are certain practical limitations to its use: when the feed chute is high and the buckets finish their loading above the level of the boot wheel there is danger of burying the wheel if the feed should be too heavy, or if the driving motor should stop by failure of electric current, or if the elevator should slow down or stop for any other reason. This is particularly true when the wheel is all the way down in the lowest position of take-up travel. High-speed grain elevators with foot wheels one-fourth as large as the head wheels are more likely to choke than those with foot wheels relatively larger, or where the speed is more moderate. Hence, unless the elevator belt is relatively wide in proportion to the projection of the buckets, so that it has plenty of pulley contact at the head, and unless plenty of power is provided to pull the buckets through an accidental accumulation of material in the boot, it is better to use a moderate speed, like those given in Table 17-B, and make the foot wheel at least half as large as the head wheel.

A moderate-speed grain elevator with a foot wheel half as large as the head wheel can use any of the standard makes of boot with rounded bottom in which the bottom of the feed chute enters at or above the level of the foot shaft in its lowest position. If the foot wheel is not so large as the ratio ½, more power is spent in stirring up the grain in the boot, and unless the feed chute is set relatively higher up, the buckets will not fill properly and the feed chute may choke.

Discharge chutes for grain elevators at the speeds of Table 17-B should be set about 30° below the center of the head shaft.

If the material is fine, dry, and dusty the speed should be kept low to give the buckets time to free themselves of air in passing through the material in the boot; otherwise they do not fill properly but stir up the material uselessly and raise an objectionable dust. Besides that, the fan action of the pulley and buckets at the head may disturb the air enough to blow the material out of the chute or waste it down the elevator legs.

For flour, bran, chaff, and similar mill products of light weight use Table 17-B up to 24-inch head wheels, but not beyond. The foot

wheel should be as large as the head wheel if possible, certainly not less than three-fourths of that size. The feed should be rather high; the lip of the discharge chute should be 20° to 25° below the center of the head wheel.

For fine cement, pulverized lime, and other dry, dusty substances weighing more than 50 pounds per cubic foot, use Table 17-B up to 42-inch head wheels, but not beyond. The ratio of head-wheel to foot-wheel diameter should not be over 4 to 3; if for any reason the diameter of the foot wheel is limited to 18 inches, for example, the head wheel must be in proportion; in this instance it should not be larger than 24 inches. The feed should be rather high; the lip of the discharge chute should be about 30° below the center of the head shaft.

If the material is hard, gritty and lumpy, like coal, ashes, coke, stone, ores, salt, fertilizers, or coarse chemicals, or if it is moist at times, as coal is, and tends to stick to the buckets, use Table 17-B. The speed of bucket travel and the number of revolutions of the head shaft are determined by the allowable wear and tear on the buckets and on the belt or chain, and not especially by the discharge of the buckets at the head. Belt elevators carrying coal crushed to 1½ inches and less are run successfully at nearly 500 feet per minute on 72-inch head wheels; but 400 feet per minute (54-inch wheels) may be considered the permissible limit for belts carrying coal larger than 2 inches, and a still lower speed, say 350 feet per minute (42-inch wheels), should be used for rougher, harder and more abrasive materials to keep wear and tear within tolerable limits. Chain elevators are never run over 350 feet per minute, and on very coarse materials 300 feet is a safer limit. If the lumps are larger than 3 inches it is better not tò use a centrifugal discharge elevator, but rather a slow-speed machine of some other type.

For ease in pick-up, and to fill the buckets properly, make the foot wheel at least two-thirds as large as the head wheel. If the space at the foot is limited, and must be made no larger than, say, 18 inches, then the head wheel should not be larger than 27 inches. Fig. 17-5 shows than when the head wheel is twice the size of the foot wheel the pressures which hinder the filling of the buckets are more than three times the forces which throw materials out of the buckets at the head. Fig. 17-1 shows how the pick-up is improved when both wheels are of the same size. In general, the larger the foot wheel, the easier the pick-up and the better the filling of the buckets. On this point see page 301.

If for a certain capacity it is necessary to run buckets at, say, 350 feet per minute, the head wheel should be 42 inches in diameter (see

Table 17-B, and the foot wheel should be at least 28 inches, 30 inches would be better, 36 inches still better. If there is room for only a 20-inch foot wheel, for instance, several things may be done: (1) make the head wheel 20 inches $\times \frac{3}{2} = 30$ inches, run the belt at 290 feet per minute (Table 17-B), and get the required capacity by a closer spacing of buckets or by using larger buckets; (2) make the head wheel larger than 30 inches and the belt speed more than 290 feet, and have an elevator wasteful of power, and with a capacity not up to normal because of buckets running only partly full; (3) use some other kind of elevator.

The boot for coal and similar coarse substances may be of any shape that will deliver the material to the sweep of the buckets if the following requirements are met: (1) it must not choke the feed inlet with dead material when the foot wheel is in its high position with the take-up all the way up; (2) the feed must not be so high that it can easily overflow or swamp the foot wheel when it is in its lowest position. For these reasons, most grain boots do not make good boots for coal, etc. The boots generally sold for this service are made with sloped fronts (see Figs. 23-9 and 23-11).

The discharge chute should be set with its lip at least 30° below the center of the head shaft; 45° is better for coal and other materials which are damp at times, and if the material is wet and fine it is well to place the chute even lower than the 45° line.

TABLE 17-C

HEAD WHEELS, SPEEDS, BUCKET SPACING, INCLINED ELEVATORS, CENTRIFUGAL DISCHARGE AT LOW SPEED

Diameter of Wheel, inches	Revolutions per minute	Belt Speed, feet per minute	Bucket Projection, inches	Bucket Spacing, inches	Diameter of Wheel, inches	Revolutions per minute	Belt Speed, feet per minute	Bucket Projection, inches	Bucket Spacing, inches
12	35	110	3	9.4	27	24	175	7	21.2
15	31	124	3½	11.8	30	23	180	8	23 5
18	28	132	4	14 1	33	22	190	9	25.9
21	27	147	5	16 5	36	21	198	10	28.2
24	25	157	6	18 8					

Table 17-C refers to elevators in which the discharge occurs without the assistance of centrifugal force. These are generally slow-speed inclined elevators, for which see Chapter 24.

Elevators for liquids can be run at the speeds of Table 17-A with buckets of the shape shown in Fig. 17-1. In order to be filled full, the buckets should complete their loading above the center of the foot shaft; or in other words, the foot shaft must be submerged. If for any reason the foot shaft must be kept above the level of the liquid in the tank, or boot, it is important to make the wheel large, first, to maintain a good depth of liquid below the shaft for the buckets to act on; second, to avoid the high pressures, almost radial in direction (see Figs. 17-3 and 17-5) which prevent the buckets from filling under the foot wheel when the latter is smaller than the head wheel.

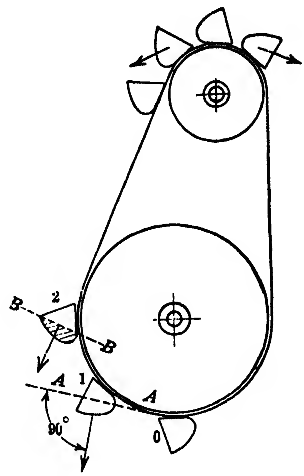


FIG. 17-9. Pick-up of Liquids in Centrifugal Discharge Elevator.

If the elevator has buckets of ordinary shape it cannot be expected to carry a volume of liquid per minute or per hour based upon the rated liquid capacity of the bucket, as stated in makers' catalogs, because, even if the foot wheel is larger than the head wheel, the directions of the resultant pressures under the foot wheel are such as to squeeze some of the liquid out of them. Buckets with a high front like Fig. 18-11 are better in that respect. Fig. 17-9 shows the pressures in an elevator run at a speed given in Table 17-A and with a foot wheel twice as large as the head wheel. Although elevators are not built with such large foot wheels, the figure is useful in showing how the filling can be improved in a high-speed elevator. Since the surface of the liquid in the bucket tends to lie at right angles to the line of resultant pressure (see page 299), the contents of a bucket at position 1 would lie parallel to *A—A*; and at 2, *B—B* represents the surface, at that instant, of what the bucket holds. Of course, the pressure due to the depth of liquid exerts a modifying influence at position 1, and the splash or the disturbed level of liquid in the boot may put more in the bucket at position 2; nevertheless it is an observed fact that if the buckets of a centrifugal discharge elevator are expected to pick up liquid below the level of the foot shaft they lift only a small fraction of their nominal capacity.

Pulps handled in the wet concentration of ores, and similar flowing mixtures of solids and liquids, behave like liquids as far as pick-up is concerned, and what is said above applies to them also; that is, for a

good pick-up the foot pulley should be larger than the head pulley. At the same time, the head pulley must not be too small. Table 17-A gives proper relations of belt speed and diameter of head pulley for high speeds. Table 17-B gives similar data for slower speeds.

If the percentage of solids in the pulp is high, and if the solids are heavy and tend to settle in the bottom of the bucket on the lift, the contents of a bucket are not discharged in a single mass, but the bulk of the water leaves first and the solids are later in leaving. This has the effect of delaying the discharge, and hence while the head receiver or chute for liquids like water or very thin pulps may be set quite high at the head of a centrifugal discharge elevator, say 15° or 25° below the center of the head shaft, the lip of the receiver should be considerably lower for heavy pulps in which the percentage of solid is high. For these 45° is often not enough; 10° or 15° additional may be necessary.

If the receiver is set low for the reason mentioned above, it is better to run the elevator at a speed given in Table 17-B; the pick-up at the lower speed will be somewhat improved, and the buckets will have a chance to take a larger load.

Delayed Discharge. The theory of elevator discharge discussed in the preceding paragraphs assumes that the material leaves the bucket with no frictional resistance. It is possible to assume a coefficient of friction between the material of which the bucket is made and the contents of the bucket and, from a known velocity and radius of rotation, calculate how far beyond the vertical a bucket will be before the tangential force overcomes the frictional resistance within the bucket. Such calculations lead to no practical result. The coefficient of friction is an uncertain quantity, and in comparison with roughnesses and dents in the buckets and the resistance offered by one or two rows of nuts and washers across the discharge opening its effect is small.

All that can be said with certainty is that, owing to the items mentioned, some of the material is delayed in discharge and some may be spilled. When the head chute of an elevator is set according to the rules given above, it will catch practically all the material, including that which is delayed in discharge. In any elevator there is always some scatter or spill which will not reach the chute, but that amount should be small and it cannot be avoided.

Rules for speeds of centrifugal discharge elevators sometimes state the travel of the chain or belt in feet per minute without reference to the size of the head wheel. *Such rules are worthless.* For proper discharge, each size of head wheel has its own best speed. Those given

in Tables 17-A and 17-B represent, respectively, the best speeds for free-flowing materials and also for those which are heavy, coarse, and not free-flowing. The figures for the number of revolutions per minute are whole numbers which differ from results of equation 2, page 297, by less than unity. Where it is not convenient to get these exact speeds by means of standard sizes of pulleys, gears, sheaves, etc., used in the transmission of power to the elevator head, a revolution more or less will make no noticeable difference in the pick-up or in the discharge.

For emphasis it may be worth while to repeat the statement that the success of a centrifugal discharge elevator may depend on the size of the foot wheel and the way material is fed to the buckets. If the foot wheel is smaller than the head wheel it may revolve so fast that the buckets pick up no material at all while they are passing around the wheel, especially if the material is fine and free-flowing. With such materials the feed should not be too low. When elevators are equipped with low-feed boots and small foot wheels, they may work, but at a reduced capacity. Sometimes they are absolute failures.

The fact that grain buckets do not take their load below or behind the foot pulleys is quite evident when a boot like that shown in Fig. 23-13, page 385, is fed at the back, where the buckets are going down. At such times, unless the front opening is closed, not only will dust come out, but the grain itself will be thrown clear out of the boot. This is especially true when, as in many grain elevators, the bottom of the front opening is lower than the upper position of the foot shaft. For the best feed from the front, the opening should be about as shown in Fig. 23-21.

Evidence from Photographs. The speeds and revolutions of head wheels given in Tables 17-A and 17-B are confirmed by successful practice and have been verified by instantaneous photographs of the discharge from the buckets. Some of these are given in Figs. 17-10, 17-11, 17-12, 17-13, which show the action of 8-by-5-inch buckets spaced 12 inches apart and running over 36-inch head wheels at various speeds. In Fig. 17-10 the wheel makes 35 r.p.m. and the buckets make a clean discharge of pebbles weighing 80 pounds per cubic foot. In Fig. 17-11 there is a clean discharge of oats at the same speed, although oats weigh only 30 pounds per cubic foot. In Fig. 17-12 the wheel makes 20 revolutions or 188 feet per minute chain speed, but the pebbles from bucket A are beginning to spill out on the chain while the discharge from B hits the inverted bottom of C. Fig. 17-13 shows a bad discharge of oats at 23 r.p.m. = 210 feet per minute.

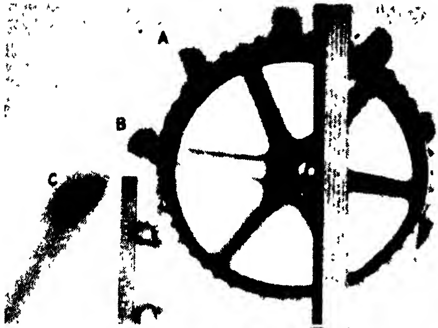


FIG. 17-10. Head Wheel, 35 r.p.m.; Chain Speed, 330 ft. per min.; Good Discharge of Pebbles.

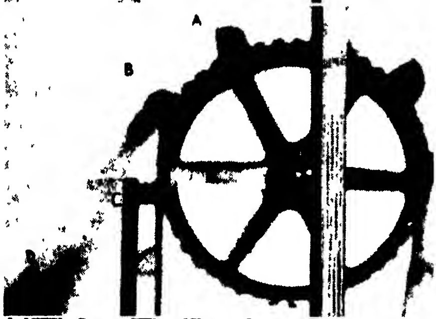


FIG. 17-11. Head Wheel, 35 r.p.m.; Chain Speed, 330 ft. per min.; Good Discharge of Oats.

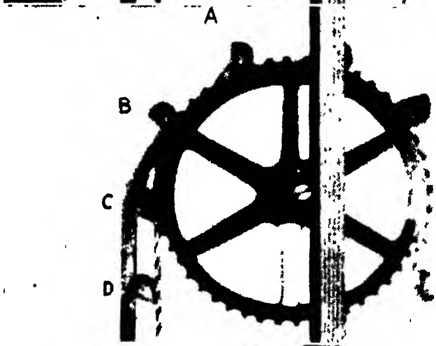


FIG. 17-12. Head Wheel, 20 r.p.m. Chain Speed, 188 ft. per min. Bad Discharge of Pebbles.

FIG. 17-13. Head Wheel, 23 r.p.m.; Chain Speed, 210 ft. per min.; Bad Discharge of Oats.

FIGS. 17-10, 17-11, 17-12, 17-13. Instantaneous Photographs of Centrifugal Discharge at Various Speeds of 36-inch Head Wheel.

Point Where Discharge Begins. In the calculations for Table 17-A it is assumed that centrifugal force equals the force of gravity, or $v^2 = gR$ (equation 1, page 297) and then discharge begins at the top of the wheel; but the lower values of Table 17-B are based on centrifugal force = $\frac{2}{3}$ the force of gravity, or $v^2 = \frac{2}{3}gR$, so that the values of v in Table 17-B are $\sqrt{0.66} = 0.82$ of those in Table 17-A. For this condition discharge will begin later and at angle from the vertical whose cosine is $\frac{2}{3}$ (see page 302). This angle is about 48°. The correctness of this reasoning is shown by Fig. 17-10, where the contents of bucket A, about 45° from the vertical, are just beginning to come out, the mass of pebbles at C being entirely clear of bucket B. Similarly in Fig. 17-11, the oats from B, which is 65° from the vertical are almost clear, the mass at D having been discharged from bucket C.

Sizes of Head Pulleys for Centrifugal Discharge Elevators. From what has gone before it is evident that three factors determine the size of head pulleys: (1) that relation between diameter and revolutions per minute which gives a clean discharge according to Tables 17-A, 17-B, and 17-C and which gives a belt speed sufficient for the quantity of material to be handled; (2) the necessity of having the diameter of the pulley at least 5 inches for each ply of the belt as discussed in Chapter 5 and Chapter 22; (3) in cases where the size of the foot wheel is limited, a ratio between diameters of head pulley and foot pulley which will permit the buckets to fill properly; see pages 306 and 309.

Fan Action at High Speeds. It is possible to run grain elevators faster than Table 17-A shows and get increased capacities, if the buckets are properly shaped and if the top of the casing is made so as to utilize the fan action of the head pulley (see page 401). A prominent maker of grain elevators set up a test plant in which the quantity elevated could be accurately measured. The normal speed of the belt was 340 feet per minute with a 28-inch head pulley. Tests run at higher speeds showed (1) that the buckets should have high end sheets, straight fronts, and rather low lips; (2) that the top of the casing must be rounded, to keep the air moving in a steady flow without swirls or eddies; (3) that casings with square tops or angular corners do not give a good discharge at the highest speeds.

When the belt was run 40 per cent faster than the standard rate, the amount of wheat elevated increased 30 per cent, of corn 40 per cent, of oats 6 per cent. At 730 feet per minute, a speed increase of 120 per cent, the corresponding figures were 60, 80, and 20 per cent; at 1100 feet, a speed increase of 220 per cent, the increases in capacity were 100, 65, and 35 per cent, respectively.

The tests showed that the fan action varied according to the weight

of the grain, also that the extra capacity was not proportionate to the increase in speed, probably because the buckets could not be filled properly at the very high speeds. From this it is evident that, since power consumption increases with the belt speed, the cost of operation, maintenance, and repair must be much higher than in a standard elevator.

CHAPTER 18

ELEVATOR BUCKETS

Discharge as Related to Shape of Buckets. In Fig. 17-1, representing the conditions of discharge in high-speed elevators, the direction of the resultant for position 7 shows that if the bucket were made with the front parallel with the back there would be interference between the front and some of the grain leaving the bucket, and that the discharge would be delayed or some grain would be trapped within the bucket. In Fig. 17-2, representing the discharge at abnormally high speed, the resultants at 5, 6, 7, 8, and 9 are all directed toward the lip of the bucket; in Fig. 17-4, showing the discharge at abnormally low speeds, the resultants all point toward the back of the bucket, and in Fig. 17-5, which represents the conditions established by Table 17-B, the resultant is parallel with the back of the bucket and there is no tendency to trap the material.

These considerations show that at the higher speeds a clean discharge is favored by having the top angle T of a bucket relatively small (Fig. 18-1) and the bottom angle B relatively large, so that the bucket presents a wide-open mouth for the release of its contents. But on the other hand, the bucket lip must not be too low, nor the bottom angle too large, or some of the material will be spilled from the bucket at position 3 (Fig. 17-1). The shapes of various styles of centrifugal discharge elevator buckets on the market represent compromises between these opposing factors. Of all the buckets for elevating grain, the

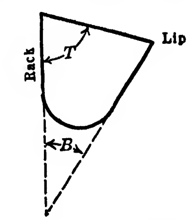


FIG. 18-1. Typical Round-bottom Bucket for Centrifugal Discharge.

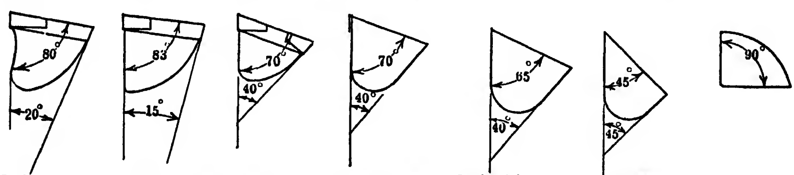


FIG. 18-2. Angles of Various Sheet-steel and Malleable-iron Elevator Buckets.

Buffalo bucket (Fig. 18-2) is most generally used for high speeds; it has a top angle of 80° and a bottom angle of 20°. Empire buckets

have a top angle 83° and a bottom angle 15°; they are large in cross section, but will not discharge at the speeds of Table 17-A. Rialto buckets with a top angle of 70° and a bottom angle of 30° or 40° will discharge at speeds even higher than those of Table 17-A, but for the same over-all dimensions they hold less than Buffalo buckets. The Minneapolis (Fig. 18-3) bucket, which is sometimes used in grain elevators, is in cross section nearly an equilateral triangle with straight sides and 60° angles; the lip is relatively low. When such a bucket on the rising side of an elevator meets the head wheel, the resultant pressure within the bucket has the direction AB, when the belt speed is standard. Then there is little or no tendency to squeeze the grain out over the lip; but if the bucket runs at the very high speeds that are sometimes recommended, the resultant has the direction AC and is greater in amount. This tends to start the discharge at the points 2, 3 (Fig. 17-2), and there may be spill and "back-legging" unless the fan action at the head (see page 311) is powerful enough to carry the grain over to the spout.

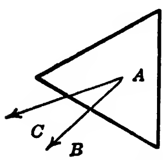


Fig. 18-3. Resultant Pressures in "Minneapolis" Bucket on Meeting the Head Pulley.

New Shape Buckets. Several recent American buckets differ from older styles in having higher end sheets and shapes similar to Fig. 18-14. The advantages of this shape have been proved by use; the high ends guide the grain in the discharge and prevent it from scattering sideways from the path of the buckets. Hence there is less spill and a greater effective capacity of the elevator. The "supercapacity" bucket resembles the Minneapolis bucket but has high ends, rounded bottom, and a greater carrying capacity; the "superior" bucket, made in two styles, is similar as to the high ends and the top and bottom angles (see page 313). These buckets are described more fully on page 318.

Malleable-iron buckets in Style A, Fig. 18-2, have a top angle 65° and a bottom angle 40° and are well suited to handle nearly all rough and heavy materials at the speeds given in Table 17-B. For materials that are wet and stringy, or which stick to the bucket, a bucket, like Style B, with a lower front will give a cleaner discharge. For material like raw sugar, which is very sticky, the wide-open mouth of the Style C bucket gives a better discharge and the material is not so likely to pack tight in the bottom.

The discharge at the head of a centrifugal discharge elevator is affected also by the shape of the bottom of the bucket; since this point is related to the spacing of the buckets it is discussed below.

Discharge as Affected by Spacing of Buckets. Fig. 17-12 shows the discharge from bucket *B* striking the bottom and front of bucket *C*, which is 1 foot ahead of it, but if the buckets were 2 feet apart there would be no such interference, because the direction of the discharge from *B* shows that the mass would clear bucket *D* 2 feet ahead of *B* even at the low speed of 180 feet per minute. At higher speeds the buckets may be closer together, and as may be seen from Fig. 17-11 the spacing for the conditions shown might be even less than 1 foot and still the discharge from bucket *B* would clear the bucket ahead of it.

A comparison of Figs. 17-1 and 17-2 shows that, as the speed increases, the material is thrown more nearly in a radial direction from the wheel and with less chance of interfering with the leading bucket. The shape of the bucket also has a bearing on the spacing; if the bottom is sharp or of small radius and lies close to the belt, the discharge from the following bucket is not so likely to strike it, and if the discharge should strike it, a straight bottom is less likely to scatter the discharge than a full, rounded bottom. This explains why buckets of the Minneapolis type (Fig. 18-6) can be placed close together, almost touching, on a belt and run at high speeds; the mouth is wide open to permit a discharge similar to that shown in Fig. 17-2, and if the discharge from one bucket does hit the bottom of the one ahead, the grain is not scattered, perhaps hardly deflected from its path.

Buckets of the Buffalo type (Fig. 18-5) have a bottom sharper and less rounded than Empire buckets and hence can be placed closer together without making a scattering discharge. Fig. 18-4 shows to scale Buffalo buckets 8 inches deep, 8 inches projection, spaced 13 inches on a grain elevator belt passing over a 96-inch head pulley that makes 27 revolutions per minute. This speed is according to Table 17-A, and the conditions of discharge are shown in Fig. 17-1. The resultants drawn from positions 6 and 7 represent the direction of the arrows shown in Fig. 18-4, and the buckets 13 inches ahead of those positions are shown dotted. It is probable that some of the discharge from 6 hits the sloping bottom of the bucket 13 inches ahead and glances off, but the discharge is most active at 7, where the resultant pressure is greater, and so directed that the grain will clear the bucket 13 inches ahead of position 7. The 13-inch spacing gives satisfactory

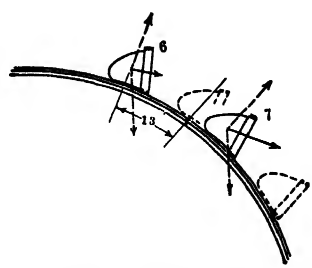


FIG. 18-4. Discharge of Grain from 15-by-8-inch Buffalo Buckets Spaced 13 Inches Apart.

results with the 8-by-8-inch Buffalo buckets in the particular elevator which Fig. 18-4 illustrates, and the amount of spill is very small.

Kinds of Elevator Buckets. Seamed Buckets. In elevators for grain, flour-mill products, etc., where the abrasion is not severe, it is customary to use light sheet-steel buckets; most of them are of the three-piece style with reinforcing band. The body sheet of tin plate or light sheet steel No. 24 or No. 26 gauge, is bent to form and fastened by lock seams to the two end pieces. The reinforcing band around the top edge, about 1 or 1½ by ⅛ inches in section, forms a digging edge in front and a hold for the bucket bolts in the back and gives the bucket the necessary stiffness to resist distortion without excessive weight. In high elevators run at high speed it is important to avoid unnecessary weight in the buckets; a heavy bucket costs more, adds to the load on the belt, and under the influence of the centrifugal forces existing in high-speed elevators it is more likely to be torn loose from the belt or pull the bolts through the belt.

Seamed buckets are sold by a number of manufacturers under trade names which are well known in this country.

The Buffalo bucket (Fig. 18-5) is made in sizes from 12 to 20 inches long, with a brace for added stiffness in the sizes over 15 inches. The back is usually curved so as to match, to some extent, the bend of the belt as it wraps around the pulley. This has some advantage in reducing the pull on the bucket bolts in picking up the load (see Fig. 21-3), but at the same time it concentrates the pressure between the back of the bucket and the belt on two spots, and belts are sometimes cut through at those places. When the buckets are made with a flat back, the wear on the belt is more distributed.

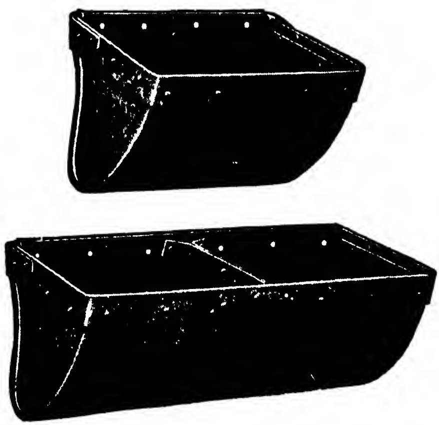


FIG. 18-5. Buffalo Buckets.

Rialto buckets are usually made in sizes up to 20 inches long; they will discharge clean at high speed, and because of their shallow form and relatively large bottom angle (see page 313) they can be placed closer together on a belt than Buffalo buckets.

Buckets of the styles known as Empire, Favorite, etc. (Fig. 18-2), are made in a great variety of sizes from 5 to 20 inches long. They

have bottoms more fully rounded than other styles and will nominally hold more for the same over-all dimensions. Because of their depth and the shape of their bottoms they must be spaced relatively farther apart to avoid interference in discharge, and hence the net carrying capacity is not greater, even under conditions favorable for fully loading the buckets. They are not often used in high-speed elevators (Table 17-A), but rather in elevators of moderate capacity where the speed is within the limits of Table 17-B. They are, therefore, well suited to handle flour, bran, chaff, and other fine, dry, light materials, as well as grain.

"Minneapolis" buckets are made in all sizes up to 20 inches. Fig. 18-6 shows the usual construction, sheet-steel body of No. 18 or No. 20 gauge with a binding strip around the top.

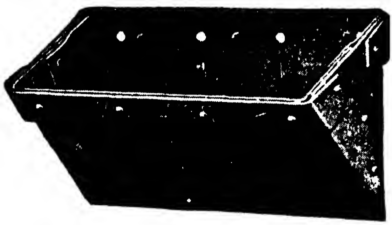


FIG. 18-6. Minneapolis Bucket.

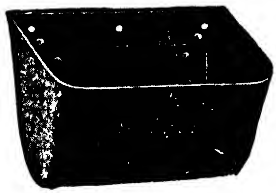


FIG. 18-7. Salem Bucket.

Pressed and Riveted Buckets. One of the oldest buckets on the market is the Salem bucket (Fig. 18-7). It is a one-piece bucket with a front nearly straight, rounded bottom, and corners riveted together or spot-welded on the back. The front, or lip, is usually not reinforced, but, in the lighter gauges, a reinforcing strip is folded over the top edge of the back. These buckets are made in sizes up to 24 by 8 inches in various thicknesses of metal. Acme buckets are like Salem buckets in shape, but the riveted or welded lap is on the ends instead of on the back.

In the lighter gauges, Salem and Acme buckets are used in moderate-speed elevators for grain and light mill products, pulverized chemicals, and similar materials; in the heavier gauges they are used for materials heavier than grain, but they will not handle coarse, heavy substances so well as malleable-iron buckets.

The Superior Bucket (see page 318) is made of a body sheet and two end sheets spot-welded together, with reinforcements at the back and lip (Fig. 18-8). The end sheets are cut high and rounded, and the bucket is rather full in the bottom, so that it holds more than most other buckets. The front is straight, the angle B (Fig. 18-1) is relatively large, and the angle T is 60° or 65°; hence, for the reasons given

on page 313, discharge begins early, and at high speeds the grain is not likely to be trapped within the bucket as it starts to descend (see page 314). The high end sheets not only improve the discharge but also enable the bucket to pick up and hold the grain better at the loading point.

The Supercapacity Bucket (Fig. 18-9) is also a three-piece bucket. It has a straight front, a full bottom, and high rounded ends. The metal is heavy enough to dispense with reinforcing; hence the back is

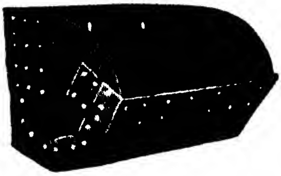


FIG. 18-8. Superior Bucket.

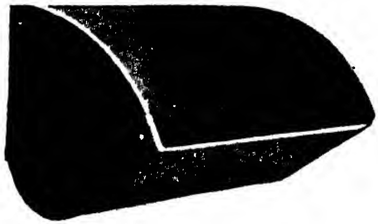


FIG. 18-9. Supercapacity Bucket.

flat and smooth and not likely to cut the belt as reinforced and banded buckets often do.

As compared with buckets of the Buffalo type, these newer buckets will carry more grain at standard speeds, and will discharge without "back-legging" at speeds higher than standard. For reasons given on page 312, it is not wise to run elevators faster than standard, but the higher speeds can be used in emergency to get more out of an existing elevator or to get an elevator of large capacity in a small space.

Seamless sheet-steel buckets, known also as Caldwell or Avery buckets, are made of one piece of soft sheet-steel pressed in dies similar in form to Salem buckets. The corners are rounded, and the bucket is stiff enough for work in grain and similar materials. However, it is more expensive than most three-piece buckets because of the heavier metal. As compared with light buckets with a reinforced top edge, seamless buckets dig their load more easily and can be run with less power.

Comparative Capacities. Table 18-A shows that the carrying capacities of various styles of grain buckets do not differ much when expressed as cubic feet per foot of belt, because the buckets of fuller cross section must be spaced farther apart to get a clean discharge.

Malleable-iron Buckets. Prior to 1908 there was no uniformity or regularity in sizes and shapes of malleable-iron buckets as made by different manufacturers. In that year the present Manufacturers'

Standard Sizes were established; the older styles and sizes have since become obsolete. The standard *A, AA, B,* and *C* buckets shown in out-

TABLE 18-A

CARRYING CAPACITY OF GRAIN ELEVATOR BUCKETS PER FOOT OF BELT

Style	Length × Projection × Depth, inches	Contents of One Bucket, cubic feet	Spacing, inches	Cubic Feet per Foot of Belt
Salem........	20 × 7 × 7	0.33	14	0.28
Buffalo	20 × 7 × 7	0.33	13	0.30
Rialto.........	20 × 7 × 6½	0.28	12	0.28
Empire..... .	20 × 7 × 7	0.36	16	0 27
Minneapolis ..	20 × 7 × 7¾	0.23	8½	0.32

line in Fig. 18-2 are illustrated also in Fig. 18-10. The bottoms are rounded to a rather large radius, and each end is flared outward at a slope of 6°; since there are no seams or rivets, the buckets fill and discharge readily and material is not likely to stick in them. The metal is

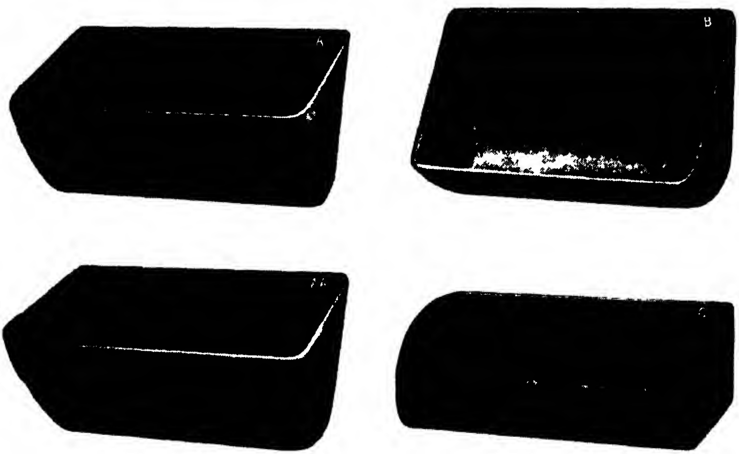


FIG. 18-10. Manufacturers' Standard Malleable Iron Buckets.

thicker than in most sheet-steel buckets; the corners are filleted and thickened; hence malleable-iron buckets are stiffer and stronger than most steel buckets of corresponding size and shape and they suffer less from distortion and abrasion in service. They are also less affected by rust.

Style A. More than 75 per cent of the malleable-iron buckets sold in this country are Manufacturers' Standard Style *A*; they are used for coal, ores, chemicals, ashes, and similar coarse materials. Table 18-B gives weights and dimensions of the regular sizes.

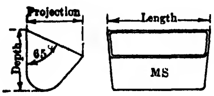


TABLE 18–B

MANUFACTURERS' STANDARD MALLEABLE IRON BUCKETS—STYLE *A*

Dimensions in inches, and average weights in pounds

Length	Projection	Depth	Capacity, cubic feet	Weight, Each
4	$2\frac{3}{4}$	3	0 01	1 0
5	$3\frac{1}{2}$	$3\frac{3}{4}$	0 02	1 5
6	4	$4\frac{1}{4}$	0 03	2 5
7	$4\frac{1}{2}$	5	0 05	3.1
8	5	$5\frac{1}{2}$	0 07	4 4
9	6	$6\frac{1}{4}$	0 11	6 1
10	6	$6\frac{1}{4}$	0 12	6.9
11	6	$6\frac{1}{4}$	0 13	7.7
12	6	$6\frac{1}{4}$	0 14	8.5
12	7	$7\frac{1}{4}$	0 19	11 0
14	7	$7\frac{1}{4}$	0 23	12 9
15	7	$7\frac{1}{4}$	0 25	13.8
16	7	$7\frac{1}{4}$	0 27	14 8
14	8	$8\frac{1}{2}$	0 30	16 7
16	8	$8\frac{1}{2}$	0 34	18 8
18	8	$8\frac{1}{2}$	0 39	21 7
20	8	$8\frac{1}{2}$	0 43	23 6
24	8	$8\frac{1}{2}$	0 51	27 5
18	10	$10\frac{1}{2}$	0.61	33 0

Style AA are like *A* buckets with the addition of metal to thicken the digging edge and the front corners. (See Table 18-C.)

Style B buckets hold less than Style *A* buckets when compared on the basis of their contained volume in cubic inches; but their carrying capacity in ordinary elevators is even less because, at the speeds of centrifugal discharge (Table 17-B), they are likely to spill a portion of their contents over the low front lip if the elevator is vertical or nearly so when the bucket meets the head wheel. In inclined elevators, when the angle is more than 30° from the vertical, there is some risk that fully loaded *A* buckets will spill over the top edge. This is not so likely to happen with *B* buckets, hence these are sometimes used in such inclined elevators. They are of greatest value in handling clay

TABLE 18-C

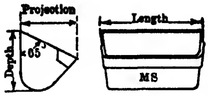


TABLE MANUFACTURERS' STANDARD MALLEABLE IRON BUCKETS—STYLE *AA*

Dimensions in inches, and average weights in pounds

Length	Projection	Depth	Capacity,* cubic feet	Weight, Each
6	4	$4\frac{1}{2}$	0.03	2.7
8	5	$5\frac{1}{2}$	0 07	4.8
10	6	$6\frac{1}{4}$	0.12	7.7
11	6	$6\frac{1}{4}$	0.13	8.6
12	6	$6\frac{1}{4}$	0.14	9.4
12	7	$7\frac{1}{4}$	0 19	12.0
14	7	$7\frac{1}{4}$	0.23	13.9
15	7	$7\frac{1}{4}$	0.25	14.8
16	7	$7\frac{1}{4}$	0 27	15.9
14	8	$8\frac{1}{2}$	0.30	18.5
16	8	$8\frac{1}{2}$	0 34	21.8
18	8	$8\frac{1}{2}$	0 39	23 5
20	8	$8\frac{1}{2}$	0 43	25.7
24	8	$8\frac{1}{2}$	0 51	30.5
18	10	$10\frac{1}{2}$	0 61	34.4

* Style *AA–RB* buckets are similar, have ribbed fronts and double-thick backs.

and stringy or sticky materials in inclined elevators. These materials often stick in *A* buckets, but discharge more readily from the wide-open mouth of *B* buckets at the comparatively low speeds used in inclined elevators.

Less than 10 per cent of the malleable-iron buckets sold in this country are Style *B*; *A* buckets are more generally useful and will carry more material per dollar of investment.

Table 18-D gives information about Style *B* buckets.

Style C buckets (see Fig. 18-2) are seldom used in belt elevators; they are suitable for wet sugar, damp clay, and similar materials too sticky to discharge from *A* or *B* buckets.

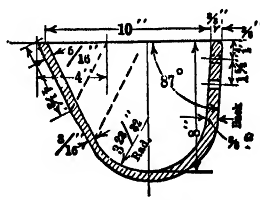


FIG. 18–11. Cross Section of 17½-by-10-inch Malleable-iron Bucket for Mineral Pulps.

Malleable-iron Buckets for Liquids and Pulps. In order to prevent loss by splashing out over the front lip at the pick-up or on meeting the head pulley (see page 307) it is an advantage in high-speed elevators to use buckets of special form with high fronts. Fig. 18-11

shows the cross-section of a $17\frac{1}{2}$-by-10-inch bucket used at a copper ore concentrating works in New Mexico. Other sizes are also made; these all have very thick backs for strength and to resist the abrasion from fine material which gets between the bucket and the belt. The fronts are also thickened. See Table 18-G.

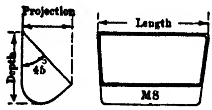


TABLE 18-D

MANUFACTURERS' STANDARD MALLEABLE IRON BUCKETS—STYLE *B*

Principal Dimensions, inches			Maximum Contents, cubic feet	Weight of Bucket, pounds (Average)
Length	Projection	Depth		
4	$1\frac{1}{2}$	$2\frac{1}{4}$	0.004	0.41
7	$3\frac{1}{2}$	5	0.03	2.2
8	$3\frac{1}{2}$	5	0.04	2.3
10	4	$5\frac{1}{2}$	0 06	4.0
12	$5\frac{1}{2}$	$7\frac{1}{2}$	0 14	6.5
16	$6\frac{1}{2}$	9	0 24	13 5

Spacing of Commercial Buckets. The parabolas 6 and 7 in Fig. 17-2 represent the discharge of free-flowing materials at the speeds given in Table 17-A, and buckets spaced as shown in Fig. 18-4 might be expected to give a clean discharge so long as the ratio of spacing to projection shown there was maintained. In practice, however, it is not advisable to space smaller buckets relatively so close as the 8-by-8-inch buckets shown in Fig. 18-4; the discharge is not quite so prompt, and so nearly radial, for several reasons. The smaller masses of material discharged from the buckets meet more air resistance, the friction between the material and the bucket walls is relatively greater, and the sheet-steel buckets generally used in the smaller sizes have bottoms somewhat fuller and more rounded than the Buffalo buckets. For these reasons, buckets smaller than 8-inch projection must be spaced relatively farther apart than Fig. 18-4 shows.

Table 18-E gives closest spacings of sheet-steel buckets for the elevator speeds given in Table 17-A. Rialto buckets can be spaced close because they are shallower than other styles in proportion to their projection and hold less material.

For the lower speeds of Table 17-B, buckets must be spaced farther apart. The reason may be seen in Figs. 17-1 and 17-2, as compared

with Fig. 17-5. At the lower speeds the discharge is more nearly tangent to the sweep of the buckets, and more clearance is needed to avoid

TABLE 18-E

FOR CENTRIFUGAL DISCHARGE ELEVATORS—CLOSEST SPACING OF BUCKETS FOR FREE-FLOWING MATERIALS AT SPEEDS GIVEN IN TABLE 17-A

Style	Projection of Bucket												
	2	2½	3	3½	4	4½	5	5½	6	6½	7	7½	8
Acme or Salem	5	6	7	8	9	10	11	12	12	13	14	15	16
Buffalo	.	.		..	...	....	....	....	12	12	13	13	13
Rialto	.	.	.	..	..	....	9	10	10	11	12	12	12
Favorite, Empire, Common Sense, Seamless	.	...	..	...	10	11	12	13	13	14	15	16	18
Minneapolis	½- to 1-inch clearance between buckets on the belt												

interference. Table 18-F gives closest spacings of buckets for the elevator speeds given in Table 17-B.

TABLE 18-F

FOR CENTRIFUGAL DISCHARGE ELEVATOR—CLOSEST SPACING OF BUCKETS FOR COARSE OR GRANULAR MATERIAL AT SPEEDS GIVEN IN TABLE 17-B

Style	Projection of Bucket													
	2	2½	3	3½	4	4½	5	5½	6	6½	7	8	9	10
Acme, Salem or any of same form. Malleable iron, *A*, *AA*, or *B*....	6	7	8	9	10	11	12	13	14	15	17	20	22	24

Pick-up as Related to Shape and Spacing of Buckets. In handling lively free-flowing material like grain the buckets can be relatively close together, because the material flows into the buckets quickly; but for coal, minerals, and coarse materials, the buckets must be farther apart to allow time for filling. The figures given in Tables 18-E and

18-F are applicable to most materials; but if the buckets of a centrifugal discharge elevator are required to handle wet, stringy, or sluggish substances, it may be necessary to space them even farther apart than Table 18-F shows. An alternative is to use an elevator of a different type, one which picks up the material at a slower speed, because vertical centrifugal discharge elevators cannot be operated successfully at speeds much lower than those of Table 17-B. Inclined elevators, perfect discharge elevators on two strands of chain, and continuous bucket elevators will discharge at very slow speeds, and for some materials they are much more efficient than centrifugal discharge elevators.

Buckets with very low fronts, like Style *B* malleable-iron buckets, will not pick up a full load at speeds higher than Table 18-F. They are more useful in slow-speed elevators of other types, especially inclined elevators.

Capacities of Belt Elevators. The capacities of elevators are usually stated as so many tons per hour, because that is the measure of the volume of material going through the plant per hour or per day, or the rate of unloading, storing, crushing, etc., in the operations with which the elevators are connected. So far as the elevating capacity of the buckets is concerned, the material lifted in one hour is sixty times the quantity lifted in one minute, but that is not always true of the operations which the elevator serves. If the supply to a crusher, mill, or machine is irregular, the feed to the elevator which takes away from the machine may for some seconds or minutes be at a rate much higher than the hourly rate which represents the working capacity of the plant. If there are under the machines hoppers or spouts large enough to store the excess quantity, they can be fitted with feeders or control gates to regulate the feed to the elevator, but most elevators are not equipped with feeding devices. Irregularity of supply to the elevator boot may come also from delays in car supply, interruptions in plant processes, and chokes in chutes, which, when relieved, put an increased burden on the elevator for a time. See also page 196.

Peak-load Capacities. For the reasons stated above, elevators should be designed for the *peak-load* or *per-minute* capacity rather than the average or per hour capacity. Neglect of this precaution leads to trouble and disappointment. An elevator seriously overloaded for less than a minute may fail for several reasons.

1. If the foot wheel is suddenly buried in material, the buckets are unable to dig their way out, the belt slips, and the elevator stops.

2. If the supply to the boot is beyond the elevating capacity of the buckets the feed chute may fill up and choke.

3. When the elevator belt slips, its speed falls off to the point at which the buckets do not make a clean discharge, the spill falls down the back leg and adds to the accumulation in the boot, and the elevator stops.

There are many belt elevators in service which are large enough to give the required number of tons per hour with regular feed and steady operation, but too small to handle the load as it is delivered to the boot *per minute*. Such elevators have trouble with chokes; belts worn, torn, or dried out by repeated slipping; pulley lagging worn off; buckets tearing off; and the annoying delays and expense of shutdowns and repairs.

Bucket Capacity and Elevator Capacity. Under favorable conditions of pick-up and discharge, the capacity of an elevator in pounds per minute equals

$$\frac{\text{Bucket capacity in pounds} \times \text{Belt speed in feet per minute}}{\text{Bucket spacing in feet}}$$

but where the bucket does not pick up a full load or discharge clean, the capacity is less. For reasons given in Chapter 23, buckets in centrifugal discharge elevators take a full load only when the belt speed and diameter of foot pulley are correctly related and when the boot is of the right shape. In many elevators these favorable conditions do not exist, the pulleys in the boot are too small, the loading is too low, and the buckets go up partly empty.

Manufacturers' catalogs rate buckets by their contents in cubic inches; this is the contained volume measured to the line AA (Fig. 18-12), but for reasons stated above, buckets in centrifugal discharge elevators do not usually fill to that line. In high-speed grain elevators when the conditions of loading and discharge are favorable it is proper to deduct 10 to 15 per cent in calculating the lifting capacity of the bucket. If the feed is such that the buckets cannot complete their loading at or above the level of the foot shaft, the deduction should be greater than 15 per cent.

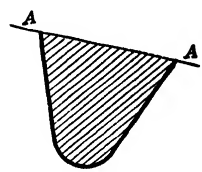


FIG. 18–12. Theoretical Filling of Elevator Bucket.

Grain elevator buckets run at the speeds of Table 17-B over good-sized foot wheels are likely to fill within 10 per cent of nominal capacity.

In ordinary centrifugal discharge elevators for coal, ores, and minerals the buckets should not be expected to carry more than 75 per cent of their rated capacity. If the feed is too low or the foot wheel too small, the capacity will not reach 75 per cent.

When inclined elevators are run at the speeds of Tables 17-B or 17-C the pick-up is good and the buckets may load to nearly full capacity if the inclination is between 20° and 30° from the vertical.

When buckets dig from an open pile, filling will not average over 50 per cent of nominal capacity.

In handling liquids or mineral pulps at the speeds of Tables 17-A or 17-B it is hardly safe to count on the buckets in vertical elevators loading more than one-third full. Inclined elevators at the speeds of Table 17-B will carry more, especially with a large foot wheel.

Examples. Buffalo buckets, 15 by 8 by 8 inches, rated, at 581 cubic inches, used in a high-speed elevator, never carried more than 14 pounds of wheat in service. This is equivalent to 537 cubic inches, a loss of 14 per cent in capacity.

In tests of a well-designed elevator for pulverized ore, with 14 by 8-inch malleable iron, Style *A* buckets, the contents averaged 400 cubic inches per bucket, a reduction of 21 per cent from the nominal capacity of 509 cubic inches.

Table 18-G gives facts about belt elevators in use in a copper smelter in New Mexico, most of them in wet service. The nominal capacities in the last line of the table are calculated on the assumption that every bucket takes a full load for every minute of the twenty-four hours. The great margin between these figures and the actual capacities in ore, over and above the water handled, represents allowances for various items: (1) some irregularity in ore supply; (2) loss of time in shutdowns; (3) imperfect filling of buckets; (4) some spill on the lift and at the discharge into the head chute; (5) using, throughout the mill, belts and buckets of standard sizes (that is, standard for that mill). This brings about a desirable uniformity among the elevators, although some are much too large for the work.

European Buckets for Flour and Grain. The ordinary "deep-pressed" bucket used in Europe is a one-piece sheet-steel bucket similar to the Salem or Acme, and has a top angle of about 70° and a bottom angle of about 33° (Fig. 18-13). The "shallow-pressed" bucket is similar in construction to the "deep-pressed" bucket, but it does not match any American bucket in shape. It has a bottom angle of about 60°, and the top angle measured from the back edge to the front lip is also about 60°; but the tops of the sides are carried up higher than the straight line which measures the top angle (Fig. 18-14). This bucket has several merits:

1. In handling flour, bran, or light mill products it will fill better because it has a wide-open mouth and the air can get out of the bucket and the material can get into it without stirring up so much dust.

TABLE 18-G

SPECIMENS OF ELEVATORS IN A CONCENTRATING WORKS IN NEW MEXICO

	Coarse Crushing Department	Coarse Crushing Department	Fine Crushing Departments				Flotation Department		
Elevator Number	1	2	3	4	5	6	7	8	9
Height of elevator	49′ 6″	59′ 6″	59′ 6″	60′ 0″	66′ 0″	66′ 0″	56′ 0″	53′ 0″	59′ 6″
Tons handled, 24 hours	6000	6000	2400	3200	200	2200	1250	250	150
Size of head shaft, diameter × length	7″×8′ $5\frac{1}{2}$″	7″×8′ $5\frac{1}{2}$″	$6\frac{1}{2}$″×8′ $9\frac{1}{2}$″	7″×7′ $11\frac{1}{2}$″	$4\frac{3}{16}$″×5′ $7\frac{3}{8}$″	7″×8′ $5\frac{1}{2}$″	7″×7′ $11\frac{1}{2}$″	$4\frac{3}{16}$″×5′ $7\frac{3}{8}$″	$4\frac{3}{16}$″×5′ $7\frac{3}{8}$″
Size of head-pulley, diameter × face, inches	60″×38″	60″×38″	60″×28″	60″×32″	40″×20″	60″×38″	60″×32″	60″×28″	60″×26″
R.p.m. of head pulley	29	32	30	28	38	30	30	28	34
Belt, width and ply	36″×11	36″×12	26″×12	30″×12	16″×10	36″×12	30″×12	26″×12	24″×12
Belt speed, feet per minute	449	510	471	431	404	460	468	438	534
Size of foot shaft, diameter × length	$3\frac{15}{16}$″×5′ $6\frac{1}{2}$″	$3\frac{15}{16}$″×5′ $6\frac{1}{2}$″	$3\frac{15}{16}$″×4′ $6\frac{1}{2}$″	$3\frac{15}{16}$″×5′ $0\frac{1}{2}$″	$3\frac{15}{16}$″×4′ $6\frac{1}{2}$″	$3\frac{15}{16}$″×5′ $6\frac{1}{2}$″	$3\frac{15}{16}$″×5′ $0\frac{1}{2}$″	$3\frac{15}{16}$″×4′ $6\frac{1}{2}$″	$3\frac{15}{16}$″×4′ $6\frac{1}{2}$″
Size of foot pulley	48″×38″	48″×38″	48″×28″	48″×32″	28″×20″	48″×38″	48″×32″	48″×28″	48″×26″
Batter or slope of belt on loaded side	24″	24″	24″	24″	24″	24″	24″	24″	24″
Buckets, length × projection	$17\frac{1}{2}$″×10″	$17\frac{1}{2}$″×10″	12″×8″	15″×9″	15″×9″	$17\frac{1}{2}$″×10″	15″×9″	12″×8″	12″×8″
Buckets, dead weight each, pounds	36	36	27	34	34	36	34	27	27
Buckets, cubic contents, each, cubic feet	0.537	0.537	0.248	0.429	0.429	0.537	0.429	0.248	0.248
Buckets, spacing on belt	18″	18″	12″	$16\frac{1}{2}$″	$16\frac{1}{2}$″	18″	$16\frac{1}{2}$″	12″	12″
Buckets, double row (*D*) or single row (*S*)	*D*	*D*	*D*	*D*	*S*	*D*	*D*	*D*	*D*
Largest size of material	3″	2″	1″	0.09″	0.02″	0.065″	0.023″	0.016″	0.008″
Ratio of water to solids as handled	Dry	Dry	$1\frac{1}{2}$ to 1	2.2 to 1	12 to 1	4 to 1	4 to 1	9 to 1	12 to 1
Size of motor	Shaft	50 hp.	50 hp.	50 hp.	15 hp.	50 hp.	50 hp.	25 hp.	25 hp.
Nominal tons, 24 hours, based on full bucket capacity	28,000	30,000	21,000	24,000	11,000	28,000	25,000	20,000	24,000

2. On account of the high ends it will carry more material than if the ends were cut off straight, as is usual in American buckets.

3. On account of the straight-line front and the large bottom angle it will discharge light mill products readily, and for the same reason there is a clean discharge of flour which tends to pack into small angles and sharp corners of buckets of some other styles. The low-front sheet-steel buckets of American makers generally have more of the front cut away; the top angle is therefore small and the bucket will discharge

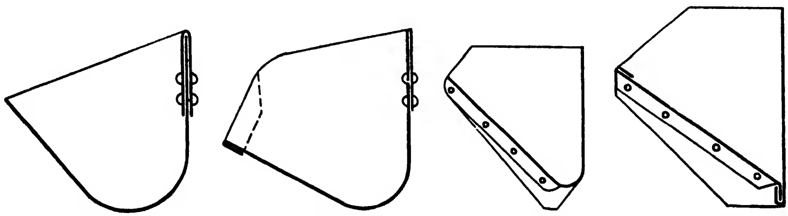


FIG. 18–13. FIG. 18–14. FIG. 18–15. FIG. 18–16.

EUROPEAN BUCKETS FOR GRAIN ELEVATORS.

FIG. 18–13. "Deep-pressed" Bucket.
FIG. 18–14. "Shallow-pressed" Bucket.
FIG. 18–15. One-piece V Bucket with Rounded Bottom.
FIG. 18–16. Two-piece V Bucket with Sharp Bottom.

readily; but the carrying capacity is unnecessarily reduced for such substances as those mentioned, for which sheet steel or tin buckets are in general use.

Buckets of angular form, like the Minneapolis bucket, are quite popular in Europe. They are made in one-piece style (Fig. 18-15) with a small fillet in the lower corner, or in two pieces with the bottom sheet separate, flanged and riveted to the ends (Fig. 18-16), in which case the bottom corner is sharp. Fig. 18-17 shows still another form of this bucket (U. S. Patent 665,273 of 1901) in which the lower edge of the front sheet is flanged and bears against the belt as a prop for the bucket. In all these, the lower edge of the front must come close to the back, so that the discharge from the following bucket will either clear, or be deflected by, the front sheet.

European makers list sheet-steel buckets with a flat back and a kind of half-conical front (Fig. 18-18). These will dig through a deep mass of grain in a boot more easily than buckets of any other shape; they are light and strong and discharge well at speeds even higher than those of Table 17-A. They can be set with little clearance on the belt and will not interfere with the discharge. A bucket of this style

10 inches wide, $7\frac{1}{4}$-inch projection, holds 0.16 cubic foot; at 10-inch spacing this is equivalent to 0.19 cubic foot per foot of belt. By comparing this with Table 18-A, page 319, and halving the figures of the

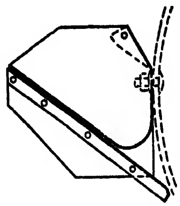


FIG. 18–17.

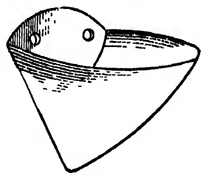


FIG. 18–18.

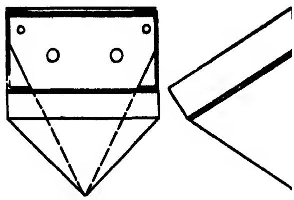


FIG. 18–19.

FIG. 18–17. European Bucket for High-speed Grain Elevator.
FIG. 18–18. "Funnel" Buckets for High-speed Grain Elevator.
FIG. 18–19. High-speed Grain Elevator Bucket, set Continuous on the Belt.

latter for a bucket 10 inches wide, it is evident that these so-called "funnel" buckets will give a large capacity.

A similar bucket having a pyramidal bottom, and made from one piece of sheet steel, is shown in Fig. 18-19. It is shown in U. S. patent 788,590 of 1905.

CHAPTER 19

CONTINUOUS BUCKET ELEVATORS

Continuous Bucket Elevators. When buckets of the shape shown in Fig. 16-2 are set close together on a belt they empty by a pouring action and do not need the throw imparted by high centrifugal force. Hence they can be run at comparatively slow speeds with merely enough velocity to dislodge the material from the bucket, to avoid dribble into the gap between the buckets, and to assist the discharge to flow promptly over the bottom of the leading bucket.

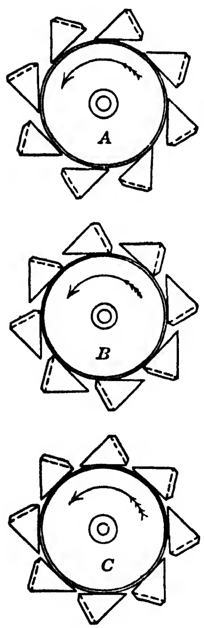


FIG. 19-1. Pick-up and Discharge of Continuous Buckets as Affected by Method of Fastening to Belt.

Pick-up and Discharge. The bolts which fasten the back of a bucket to a belt must necessarily be in one or, at most, two rows so as to allow the belt to bend on the pulleys (see page 350). In fastening the comparatively shallow and round-bottomed buckets used in centrifugal discharge elevators, the bolts are near the top of the back edge, but, with the deeper and heavier buckets used in continuous bucket elevators, the bolts must be about halfway between top and bottom in the back plate. In Fig. 19-1 the top half of diagram *A* represents continuous buckets bolted near the top edge, passing over the top of a pulley, while the bottom half shows the action under a foot pulley. The pick-up under the pulley might be called good, but the discharge is bad, because material will be poured out of one bucket into the space between the belt and the bucket ahead. In diagram *B*, showing buckets bolted at the bottom, the pick-up is bad, because material will enter the gap between belt and bucket, but the discharge is good. If the buckets are fastened at the middle, according to standard practice (see diagram *C*), the result is a compromise; the gap which opens between the belt and bucket is relatively small and at the discharge

point material is not likely to get into it unless the speed is so slow that the contents of the bucket dribble out. Material would get into the gap if the bucket picked up its load from a boot; for that reason, belt elevators of this kind should never be loaded from a boot, but from a chute (Fig. 19-2) at a point above the foot pulley where the buckets are on the straight run, lie close to the belt, and do not present any gap into which material might enter. When material does get between the belt and the back of the bucket it may wedge there tightly, putting a severe strain on the bolt fastening, and perhaps wearing or punching holes in the belt from repeated pressure in going over the pulleys. See page 351.

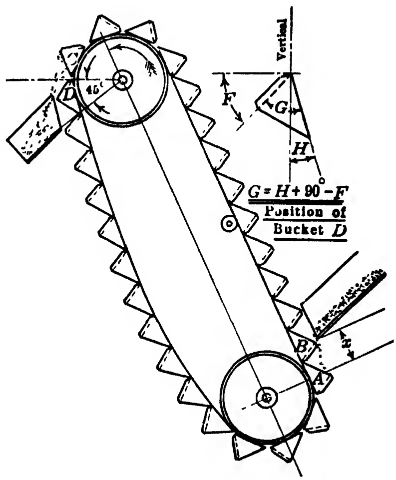


FIG. 19-2. Loading and Discharging Continuous Bucket Elevator.

The distance x (Fig. 19-2) from the lower edge of the loading chute to the shaft in its upper position of take-up travel should be at least equal to the height of one bucket; more is preferable, to make sure that the bucket A will catch all that B misses, and that A is flat against the belt when the feed pours into it. At the head, the upper edge of the discharge chute is usually set at 45° below the level of the head shaft and as close to the elevator as the sway and movement of the buckets will permit; then if the buckets are properly shaped and run at proper speed, stone, coal, gravel, and such materials handled in continuous bucket elevators will flow out in a clean discharge. If the material, gravel, for instance, is quite wet and contains sand, it is better to put the chute lower by a foot or two to catch the delayed discharge and the drip.

Shape of Continuous Buckets. In Fig. 19-2 the contents of C are pouring out over the bottom of D, and it is evident that a clean and prompt delivery to the chute depends, among other things, on the angle at which the bottom of D stands at the moment it acts as a chute. In most continuous bucket elevators there is some "throw" which helps material across D to the chute, and some of the discharge may enter the chute without touching D at all; nevertheless, there is always some spill onto the bottom of D which must slide off.

In order to make the angle F (Fig. 19-2) steep enough to let material slide easily, the bucket must have the right shape; the angle G

in the bottom must bear a proper relation to the material handled and to the angle H representing the deflection of the bucket from the vertical at the position D. If the elevator stands vertical H is zero and $G = 90° - F$; if, for instance, the material is clean and fairly dry, it will flow on a steel plate when $F = 40°$; then the bucket should have a bottom angle of 50°. It is generally better to have G larger than 50°, so that material will not be so likely to wedge in the bottom corner, and since gravel, damp coal, stone, dust, and similar substances flow more readily when F is 45°, it is an advantage to incline the elevator so that H is at least 10°. For this condition $G = H + 90° - F$ or $10° + 90° - 45° = 55°$. If the elevator has a greater inclination from the vertical, H might be 15°, then the buckets could have a bottom angle G of 60°. The angle H can be taken from Table 24-A; strictly considered, the angles B in that table refer to belts without applied tension; when take-up tension is applied, B becomes less and H greater, but for practical purposes, the table may be followed and H may be considered equal to $90° - B$.

On the basis of these considerations, as well as those referring to the loading, most continuous bucket elevators on belt are inclined from 15° to 30° from the vertical and have buckets with the bottom angle G from 50° to 60°. Some buckets are made with G equal to 70°, but they do not give a clean discharge unless the elevator is inclined at least 25° from the vertical and handle dry, clean, free-flowing material. For that case, H is about 10° or 12° and $F = 30°$ or 32°.

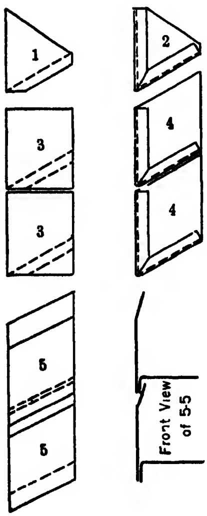


FIG. 19-3. Some Forms of Continuous Buckets.

From a consideration of the way the feed enters the bucket from the loading chute (Fig. 19-2) it is evident that the buckets must be open in front. They would hold more if made with a short front sheet parallel to the back sheet, but such a front would interfere with the loading, would splash and spill the material, and would soon wear out.

Fig. 19-3 shows various shapes of continuous buckets, No. 1 being the ordinary two-piece bucket, one sheet forming the back and ends, another sheet, flanged at each end, forming the bottom. The corresponding three-piece bucket, No. 2, has two end pieces riveted to one sheet which makes the back and bottom. In order to prevent scatter and spill at the loading chute, buckets are sometimes made like No. 3 or 4, but they have given trouble at times, by stones catching between

the end plates of adjacent buckets, and the projecting corners of the end plates may be knocked out of shape in handling heavy material. These mishaps are not likely to happen with buckets like Nos. 1 and 2, because the end plates do not butt against each other and the corners have been trimmed off. Buckets like No. 5 have been made with the idea of improving the discharge, as where C discharges onto the back of D (Fig. 19-2). The overlapping end plates confine the discharge and prevent it from scattering, but in the course of time the overhanging corners of the plates are likely to get out of shape and then spoil the overlap, or interfere with each other. Buckets, in end view like No. 2 or 4, are sometimes made tapering, narrower at the top than at the bottom so as to overlap and thus prevent the discharge from spreading out too far sideways. They have some advantage in that respect, but, on the other hand, loading is not so likely to be clean as when the end plates of the buckets are all in one plane, that is, not tapered.

Height of Bucket and Diameter of Pulley. In order that the bucket may not gap away from the belt too far (see diagram C, Fig. 19-1) the bucket must not be too high, measured along the belt, nor must the pulley be too small. Table 19-A gives the amount of the gap in inches,

TABLE 19-A

RELATION OF HEIGHT OF BUCKET TO DIAMETER OF PULLEY

Height of Bucket, inches	Gap Between Belt and Bucket, inches — Diameter of Pulley, inches					
	18	24	30	36	42	48
8	0.84	0 64	0 52	0 43	0.37	0.33
10	1 29	1 00	0 81	0.68	0.58	0.51
12	1.81	1 41	1.15	0 97	0.84	0.73
14	2.40	1.82	1.55	1.31	1.13	1.00
16	3.04	2.42	2 00	1 69	1.47	1.29
18	3.72	3.00	2 49	2.12	1 84	1.63

measured radially, for various heights of buckets and diameters of pulleys. The gap should never be more than one-eighth the height of the bucket, and therefore combinations to the left of the heavy line in the table should not be used.

Buckets more than 16 inches high are seldom used, even with large

pulleys; it is hard to fasten them securely to the belt with two or even three rows of bolts.

Capacities of Continuous Buckets. It is not possible to give rules for the carrying capacities of continuous buckets. The loading chute is always narrower than the bucket, and the ends do not fill so well as the middle of the bucket. The amount that can be piled in a bucket depends on the piling angle of the material, the shape of the front of the bucket, and the inclination of the elevator; but it does not usually exceed 75 per cent of the maximum represented by the cubic contents of the bucket. It is generally necessary to make a sketch to determine the carrying capacity.

Speeds of Continuous Bucket Elevators. On the use of continuous buckets for elevating grain at high speed, see page 329. Generally the term "continuous bucket elevator" is applied to machines which run at speeds lower than those of Table 17-B and which do not depend altogether on centrifugal force to empty the buckets. The low limit of speed is that which will prevent material from dribbling into the gap between the buckets (see page 332), and the high limit is determined by the nature of the material and the delivery to the buckets. Such elevators on belt are seldom, if ever, used to pick up material from a boot because of the danger that material will crowd between the belt and the bucket, pack there, and pull the bolts through the belt or tear the buckets off. When they are fed from a chute at some distance above the foot wheel, as they should be (see page 331), the considerations are that the bucket should have sufficient time to fill properly, and that the impact of the material delivered to the bucket should not be too severe for the fastening to the belt. If the material is large in proportion to the area of cross section of the bucket, or if the pieces are long and flat, like shale or some kinds of crushed cement rock, the load does not settle quickly into position in the bucket and a relatively slow speed is necessary to load the bucket properly and avoid spill over the front edge. Elevators that handle such material may be run between 100 and 150 feet per minute.

Speed Must Not Be Too Low. In handling materials, like moist fine ores or excavated earth or other substances which do not flow readily on themselves, continuous buckets may fill with a high surcharge. In passing over the head pulley the surcharge will spill into the gap between the buckets, unless the speed is high enough to influence the discharge by centrifugal action. For such conditions it is advisable to incline the belt or use protectors like Fig. 19-4.

Objections to High Speed. If the material is small, like crushed slag or stone for road building, railroad ballast, or concrete construc-

tion, the speed may be higher without much risk of spill and scant loading. Two hundred feet per minute has been considered standard for such work, but there are many belt elevators running at higher speeds which are thought to be satisfactory. In some cases the belt travels as fast as 300 feet per minute; the object, of course, is to get a high capacity from a belt and bucket of given size. Some of these elevators handle small stuff, relatively light in weight, like crushed slag, satisfactorily; but in others carrying heavy, coarse materials the high speed causes an excessive amount of spill and unusual wear and tear on the belt and buckets. Buckets 15 inches high traveling 300 feet per minute pass a loading chute at the rate of 4 per second. Considering the depth of material in the loading chute, the time of loading such a bucket is perhaps as much as one-third of a second, certainly no more. This may be sufficient for fine material to run into and fill a bucket, but not so with coarse stuff. There are many elevators handling stone 2 inches and larger where the buckets do not take a full load and where the capacity elevated would be greater if the belt speed were less. Lower speed means better filling, less spill and scatter into the pit at the foot of the elevator, and less pull on the bucket bolts at the loading point and in going under the foot wheel and over the head wheel.

Sizes of Pulleys. For reasons given in Chapters 5 and 22 the diameter of head pulleys should be at least 5 inches per ply of belt. Pulleys smaller than this do not grip the belt so well, and the belt wears out sooner because of slip or by reason of excessive stress on the friction rubber or the stitching which holds the plies together.

Loading Legs for Continuous Bucket Elevators. In order to reduce the amount of spill and scatter, continuous buckets are sometimes run between side boards or plates at the loading point, or the two sides may be joined by a front sheet below the loading chute so as to form a three-sided box a few feet deep. In some cases these have worked well, but in others they have been tried and then thrown out. In stone elevators, the material is likely to get into the clearance spaces between the moving buckets and the fixed plates, and either wear out the plates or damage the buckets and the belt. The fastening of a continuous bucket to a belt is necessarily confined to a number of bolts that perforate a narrow strip across the width of the belt; anything that puts an added strain on this section of the belt is to be avoided. If the loading chute is properly sloped and set with reference to the buckets, and if the belt speed is not too great, the amount of spill is ordinarily not objectionable. Usually it is better to clear away the spill regularly rather than try to prevent it by the use of a loading leg.

For further information *about inclined continuous bucket elevators* see Chapter 24. Refer also to Chapters 20 and 22.

Belt Protectors for Continuous Bucket Elevators. The Stephens belt protector (Fig. 19-4), made by the United States Rubber Company, is designed to prevent material from getting into the gap which opens between bucket and belt when the belt bends on a pulley. It consists of a strip of heavy fabric, rubber-covered on both sides. One

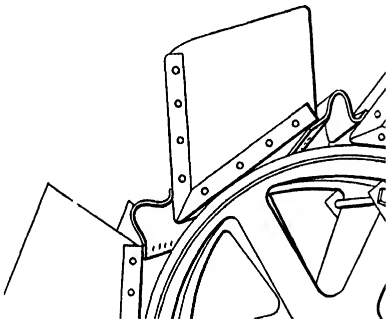


FIG. 19-4. Stephens Patent Belt Protector.

edge is riveted to the top of one bucket and the other edge to the bottom of the bucket ahead, with enough slack to allow the buckets to move with relation to each other.

Another device used in the Pittsburgh district is a strip of 3- or 4-ply rubber belt with one edge riveted to the top edge of one bucket and 3 or 4 inches of its width bearing against the bottom of the bucket ahead, but not fastened to it. Here the flexibility of the strip allows for the motion of the buckets.

CHAPTER 20

BELTS FOR ELEVATORS

Belts for Elevators. As far as materials of construction and methods of manufacture are concerned, elevator belts are like conveyor belts, and what is said in Chapter 3 applies generally to belts for elevating as well as for conveying.

Elevator service may be considered an extension of conveyor service; a belt will convey certain material on the level or on any slope up to 20°; fitted with cleats to prevent the material from rolling or sliding, it will carry up to, say, 30°; fitted with buckets to hold the material it will carry at any angle up to the vertical.

There are some features of elevator service which make that work harder for belts than conveyor service:

1. Most elevators are under 75-foot centers, very few are over 150-foot centers; they are shorter than conveyors, and since the speeds are not much different, the belt makes more contacts with the pulleys and is bent oftener.

2. Elevator pulleys, especially foot pulleys, are smaller, considering the number of belt plies, than corresponding pulleys on conveyors; the tendency to stretch or break the bond between the plies of fabric is therefore greater.

3. The unit stress in the belt, that is, the pounds pull per inch per ply, is often greater in an elevator belt than in a conveyor belt (see page 294).

4. Elevator belts are subject to cutting and wear on the outer side from material delivered against it by feed chutes or in the boot; they are cut by the top edge of the back of the bucket and worn by bits of material caught between the belt and the bucket. On the pulley side they are often gouged and torn by the heads of the bucket bolts or by material falling between the belt and the foot pulley. Belt creep and belt slip wear the belt (see page 372) by rubbing on the head pulley; the same thing happens when dirt and grit adhere to the pulley side of the belt; water enters through the bolt holes and destroys the cotton fiber and the bond between the layers of fabric.

5. A conveyor does its work in the open; the manner of loading can

be seen and the condition of the belt observed. An elevator is usually enclosed and the pick-up is not visible. It often happens that loose buckets and loose bolts are not noticed in time to prevent damage to the belt.

6. An overloaded conveyor belt gives warning by spilling the excess over the sides where it can be seen. A choke in an elevator boot is usually hidden from view; it may cause the belt to slow down or stop, and before the drive can be stopped, the head pulley, continuing to turn, may damage the belt or wear it through.

7. Elevator belts are more easily overloaded than conveyor belts. The normal ratings of conveyor belts are much less than the maximum loading (see Chapter 8). Elevator capacities are often calculated from the cubic capacity of the buckets as given in manufacturers' catalogs. These may be realized if the pick-up is good, but where, as often happens, the foot wheel is too small or the boot is not suited to the material handled, the buckets do not take a full load. It is a matter of observation that an elevator with a steady feed to the boot will run with a series of buckets only partly full; as the accumulation in the front of the boot piles up to the center of the foot wheel or higher, the buckets fill full for some seconds; then the loading falls off again and the cycle is repeated. The result is that the capacity of the elevator is less than was calculated, and the margin between the actual and the calculated capacity may be so little that a slight excess of feed may choke the elevator, with resulting injury to the belt.

The first belt elevators were used for grain in flour mills; their use has now been extended to practically all kinds of elevating work except dredging.

Grain Elevator Belts. Oliver Evans, in his "Miller's Guide," published in Philadelphia in 1795, describes as one of his inventions, what was then a new thing, at least to the milling business, i.e., an elevator composed of a leather belt with cups or buckets fastened on at intervals and discharging their contents by *centrifugal action* over the upper pulley. To handle 300 bushels per hour, the elevator consisted of a strap of harness leather 4½ inches wide with buckets holding 1 quart strapped onto the belt every 15 inches. The head pulley was 24 inches in diameter and made about 30 revolutions per minute. The foot pulley was smaller and was contained in a wooden boot which was part of a wooden casing with double legs. The buckets were of willow wood ⅜ inch thick, steamed and bent to form the front and ends; a piece of leather tacked on formed the bottom, and the elevator belt acted as the back, the elevator being inclined 15° or 20° from the vertical. Evans showed also how to make buckets of sheet iron, but that material was

scarcer than willow wood in the United States in 1785, when the first of these elevators was built.

Evans and his successors, Ellicott and others, built many belt elevators in flour mills up to 1830; in 1842 when Joseph Dart built the first bulk storehouse ("elevator") for grain on the Great Lakes at Buffalo, it had a belt elevator capable of handling 1000 bushels per hour. By 1866 many "elevators" had been built at Cleveland, Toledo, Chicago, Milwaukee, and other Lake ports, some of them holding over a million bushels. The first "elevator" on the Atlantic coast was erected in Philadelphia between 1859 and 1863; it had elevators with leather belts 20 inches wide and ½ inch thick. Rubber belts came in about 1870 and became popular during the rapid building of grain "elevators" in this country during the seventies and eighties. Some of these belts were of excellent quality (see page 52), but as to their construction, the specifications for the early elevators give little information; those for the Pennsylvania Railroad Company's Girard Point Elevator in Philadelphia in 1881 merely call for "best quality smooth-surface gum belts."

With the growth of the business came the detailed specification for rubber belts, such as the Metcalf specification (see Chapter 3). This specification and others similar to it served a purpose, but, there is a growing dependence upon the quality which experienced manufacturers have put into their trade-marked belts and less insistence upon some of the details of the older specifications. What the purchaser really wants in a grain elevator is a belt that will last many years without separation of the plies; it is not possible to get this by a specification that merely calls for so many pounds friction test, because a high friction test does not necessarily mean a rubber that will last a long time before it loses its elasticity and tenacity (see page 53).

Rubber belts for grain elevators, as now made, are generally of 32-ounce duck, 5, 6 or 7 plies thick according to the service. In general it is safe to use a "friction surface" belt, that is, one which has only the thin layer of friction rubber on the outside surface (Fig. 3-3), but where the grain is handled wet, as in oats bleachers, a rubber cover $\frac{1}{32}$ or $\frac{1}{16}$ inch thick all over is necessary. In the past, rubber-covered belts were standard for all grain elevator legs, but since the work is dry and not abrasive, a rubber cover is not essential. Some experienced buyers prefer to spend their money for quality of friction rather than for rubber covers on leg belts. The fact is that rubber belts with a low-grade friction do not hold together unless they have the protection of a rubber cover. The friction in "competition" belts is likely to have a high percentage of mineral matter, and consequently a poor bond

between the plies of duck; with these belts, the standard rubber cover, which is $\frac{1}{32}$ or $\frac{1}{40}$ inch thick, serves a useful purpose in keeping out atmospheric moisture; and it prolongs their life. In belts of better grade, the friction is compounded to maintain its elasticity for a long time; it forms a good bond between the plies of duck and does not need the protection of a cover in the ordinary work of elevating grain.

The trade-marked belts made for grain elevator work by experienced manufacturers are built of 28- or 32-ounce duck with an 800- to 1000-pound cover, if the belt has one. First-grade belts show 16 to 19 pounds friction test; second-grade belts show 12 to 15 pounds test.

Rubber Belts for Other Service. Many good conveyor belts are made from 28-ounce duck, but elevator belts are seldom made from duck lighter than 32 ounces. A standard 32-ounce duck may have in the warp 23 threads per inch, 7 yarns per thread, and in the filler 13 threads per inch, 6 yarns per thread. Belts for heavy service may have 34-ounce or 36-ounce duck; the heaviest belts are built from 42-ounce duck.

As has been stated (page 53), the weight of duck is not in itself a measure of the strength or worth of the belt; those qualities depend also upon the proportion of warp and filler threads, the twist of the threads, and the manner in which the plies are held together by the friction compound. The skill and knowledge of the belt manufacturer in combining these with the proper grade of friction and with the right covers, when necessary, determine the value of a belt for a particular service and its ability to withstand, to an economical degree, all those strains, shocks, cuts, punctures, and other distresses to which elevator belts are liable.

Friction-surface belts have, on their outer surfaces, only that thin layer of friction rubber which is calendered or pressed into the duck before it is assembled and cured. They can be used economically for some fine dry materials like crushed ores, pulverized dry coal, and dry chemicals; but where the material is damp, as coal often is from exposure to rain or snow, or where the ore is wet, as from jigging or as handled in the flotation process of ore separation, then a rubber-covered belt is preferable for several reasons:

1. The rubber cover keeps moisture from penetrating the cotton fabric.

2. It forms a cushion to prevent the fine particles which stick to the belt and pulleys from being pressed into the fabric.

3. When the belt slips and creeps on the head pulley as it always does (see page 372) the rubber protects the fabric from being worn away by the grit always present between the belt and the pulley.

Wear on elevator belts is internal and external. If the material handled is clean, dry, not abrasive, and not lumpy, belts may fail because the friction dries out and the plies come apart; if the material is clean and wet, the plies separate sooner because water enters through the bolt holes or through cuts and cracks; and the external wear is also a factor because the belt is more likely to slip on the wet pulleys. If the material is sharp and cutting as well as wet, the external wear is more rapid, and if the pieces are also hard and large, internal and external wear are both more serious from the cuts and punctures which the belt receives.

Adding covers to elevator belts is a means of equalizing the external and internal wear; it produces a balanced construction which prolongs the life of the belts. In that respect it is like adding covers to conveyor belts. Rubber belts were not a success for conveying coarse materials heavier than grain until Robins made them with rubber covers. Similarly, in many elevators handling ore, and especially wet ore, the cost of upkeep has been greatly reduced by the use of belts with rubber covers on one or both sides. In other words, in spite of the greater cost of belts with rubber covers, the cost of elevating one ton of material has been reduced and less time has been lost in shutdowns for repairs and replacements of belts.

Rubber-covered Belts. The ordinary $1/32$ or $1/40$ inch of rubber which characterizes the cheapest rubber-covered belt serves its purpose when it is required merely to keep out atmospheric moisture, when the material is not lumpy or abrasive, and when the slip of the belt on the head pulley is comparatively slight. Where conditions are bad in these respects, a thicker cover is needed to make a balanced construction. On the pulley side, a rubber cover maintains a good contact with the head pulley in spite of dirt and grit; if the work is dry, it increases the coefficient of belt contact; if the work is wet, the coefficient may not be any greater than between a wet pulley rim and a friction-surface belt, but the cover certainly acts as a protection to the fabric when the belt creeps and slips. A cover on the pulley side acts also as a cushion to prevent injury to the fabric from hard pieces jammed between the belt and the foot pulley.

It also forms a cushion into which the heads of the bucket bolts can sink without tearing the fabric, and the heads are not so likely to come into contact with the pulley rim. When the bolt heads project beyond the belt surface and bear against an iron pulley rim, the belt tends to slip, especially if wet. If the head pulley is lagged, the lagging may be cut and torn by the bucket bolts. Conversely, the cover on the pulley side of the belt protects the fabric from being cut by the lagging bolts

or rivets which often project when the lagging wears thin. This may occur from the natural creep of the belt even though the belt may apparently not slip (see Fig. 22-1 and page 373).

Fig. 20-1 shows a group of bolts used to fasten rubber lagging to the rim of a head pulley. The bolt heads have been worn away by the slip and creep of the elevator belt.

A rubber cover on the bucket side of the belt helps in several ways.

1. It protects the fabric from wear caused by direct impact of material against the belt in the boot or from a feed chute.

FIG. 20-1. Lagging Bolts Worn by Slip and Creep of Elevator Belt.

2. It resists the tendency of the forward edge of the bucket to cut the belt, either from hanging forward on the down run or from the action of centrifugal force in passing around the wheels.

3. It prevents the bits of material which catch back of the bucket from being forced into the fabric.

Figs. 3-8 and 3-9, page 42, show two 8-ply elevator belts, one with the ordinary $\frac{1}{32}$-inch rubber covers on each side; the other has $\frac{1}{32}$ inch on the pulley side, $\frac{1}{8}$ inch on the bucket side, and a protection for the edge made by carrying the top cover around to the under side.

Thickness of rubber covers on elevator belts has been a matter of trial and investigation for several years. The size of the ore, its sharpness, whether it is wet or dry, the size of pulleys used, the size and spacing of buckets—all these are factors which determine the best proportion of belt plies and belt covers which makes a balanced construction for a particular elevator. These factors differ in various plants, and therefore the belt specification for the best service and lowest cost of handling will also differ.

Examples of current practice are given below:

1. An elevator 67-foot centers for cement clinker, size 1 inch and under, 60 pounds per cubic foot, temperature 160° F. Steel buckets 20 by 12 by 8 inches every 17 inches on the belt. Head pulley 48 inches, foot pulley 36 inches. Belt 22 inch, 8-ply, 36-ounce duck—the manufacturer's best grade, cover on pulley side $\frac{1}{32}$ inch, cover on bucket side, $\frac{1}{16}$ inch. Speed 400 feet

per minute = 32 r.p.m. of head pulley. Belt joint lapped and bolted. Belt lasted 450 days, working 24 hours a day. Life and service considered very satisfactory.

2. An elevator 57-foot centers, inclined about 10° for lead ore, size ⅜ inch, wet, 100 pounds per cubic foot. Steel buckets 14 by 7 by 5½ inches every 22 inches. Head pulley 42 inches, foot pulley 30 inches. Belt 16-inch, 8-ply, 36-ounce duck—the manufacturer's best grade, cover on pulley side 1/32 inch, on other side 1/16 inch. Speed 385 feet per minute = 35 r.p.m. of head wheel. Belt fastened with Jackson fastener. Belt lasted 682 days, elevated 455,000 tons; cost of belt per ton of ore 0.076 cent. This is a very good record.

3. An elevator 48 foot-centers, inclined about 15°, for rejections from jigging ore, size ⅜ inch and under, damp, 160 pounds per cubic foot. Steel buckets 22 by 7 by 7 inches, spacing not stated. Head pulley 31 inches, foot pulley 24 inches. Belt 22-inch, 7-ply, 36-ounce duck—the manufacturer's best grade but not rubber covered (friction surface). Speed 520 feet per minute = 64 r.p.m. of head wheel. Belt lasted 126 days; cost per ton of material elevated 0.14 cent. This was better than the record of other belts used in this elevator, but the life was not so long as it might have been had the belt been rubber-covered and run at a slower speed. Five hundred and twenty feet per minute means 83 r.p.m. of the 24-inch foot wheel—entirely too fast for the pick-up of such material (see page 298). The pulleys were too small in diameter for 7-ply belt (see Chapter 5).

4. An elevator 55-foot centers, inclined slightly, for wet ore, ½ inch size, 100 pounds per cubic foot. Steel buckets 14 by 7 by 5½ inches spaced 18 inches, head and foot pulleys 36 inches. Belt 15-inch, 6-ply, grade not stated; covers both sides 1/16 inch, belt joint, Jackson fastener, belt speed 470 feet per minute = 50 r.p.m. of head pulley. Belt lasted 632 days, 24 hours per day. A good record; belts seldom lasted over a year in this elevator. Speed is too high for a 36-inch head wheel.

5. An elevator 58-foot centers, inclined about 10°, for wet ore, size ⅜ inch, 100 pounds per cubic foot, steel buckets 14 by 7 by 5½ inches every 18 inches, head and foot pulleys 36 inches. Belt 15-inch, 7-ply, 36-ounce duck—the manufacturer's best grade, rubber-covered, thickness not stated, speed 470 feet per minute = 50 r.p.m. of head pulley. Belt lasted 416 days, handled 436,000 tons; belt cost per ton elevated 0.08 cent. Considered a good record; the user expects belts to last 10 or 12 months on this elevator. Speed is too high for a 36-inch head wheel.

6. An elevator 65-foot centers, inclined about 25°, for crushed stone, size 3 to 8 inches, dry, 100 pounds per cubic foot. Buckets 34 by 16 by 14 inches continuous on belt. Head pulley 48 inches, foot pulley 48 inches. Speed 250 feet per minute = 20 r.p.m. of head wheel.* Belt 38-inch, 10-ply, 36-ounce duck—the manufacturer's best grade friction-surface belt, no rubber covers. Lasted 4 years 8 months and carried between 1,500,000 and 2,000,000 tons of stone. An excellent, the best ever made on that elevator.

* This is not a centrifugal discharge elevator.

Choice of Elevator Belts. The kind of belt best suited to a particular elevator can be guessed at from a knowledge of what has given good service under similar conditions elsewhere; but since operating conditions are never exactly alike, the question can be settled in a purchaser's mind only by trial. The true test of an elevator belt is, "What does it cost per year or per ton of material elevated?" There are places where a belt is always discarded on account of external injuries, which cannot be avoided without serious changes in methods or equipment and at too great expense, and in which the wear cannot be resisted by any degree of high quality in the friction, or duck, or cover of the belt. In such cases, to buy expensive belts is throwing money away. On this subject, see Chapter 3.

In normal elevators, however, the ordinary causes of belt failure are well known, and they can be opposed successfully by using the right kind of duck or the proper kind of friction or covers suited to the work. The main thing is to have the belt so strong that it will not fail suddenly under an accidental overload, or a choke from buckets ripping off, or a stick falling into the boot, etc. A sudden shutdown through the breakage of a belt may cost more in lost output of product, and in emergency repairs, than the price of a good belt. When the right belt has been chosen, it should be inspected regularly, and it will give plenty of notice before it is worn out; then a new belt can be ordered at the proper time in advance, and when the day comes to put it in place, the replacement can be made in an orderly routine way without interruption of service and at the least cost.

On the relation between bucket width and belt width, see page 355.

On the relation between belt tension and belt thickness, see page 326.

Wet Elevating. Handling the semi-liquid pulps in the wet concentration of ores is hard work for belts. Ordinary belts do not last long; the coefficient of belt contact with wet surfaces, rubber to rubber or rubber to iron, is only about half that of a dry belt on a lagged pulley; the belt is likely to slip unless it is pulled tight, and that slip in the presence of the fine grit which sticks to belt and pulleys wears out the driving face of the belt. The fine sand in the pulp works into the fabric through cuts and cracks, the plies separate, and sand blisters form. The water, getting into the cotton mildews it, and the plies come apart. For these reasons a friction-surface belt is out of place in handling mineral pulps, and even an ordinary rubber-covered belt with its $\frac{1}{32}$ or $\frac{1}{40}$ inch of cover does not usually return good service for the money spent on it, because the thin rubber is soon rubbed off by the slip and creep of the belt combined with the gritty sand.

Covers for Wet Elevator Belts. Experiments and tests under working conditions made by mining companies have demonstrated the value of comparatively thick covers on the pulley side of belts handling wet ores and mineral pulps. One copper company uses belts with 5/32-inch cover on the pulley side and 3/32-inch on the bucket side; another company uses covers 3/16-inch and 3/32-inch on pulley side and bucket side, respectively; a lead-mining company uses 3/16-inch rubber on the pulley side and only 1/32-inch on the other side, depending on flaps of old belt (see Fig. 21-2), bolted under the buckets to protect the belt from cutting and abrasion.

Fig. 20-2 shows cross sections of elevator belts made especially for handling wet pulps (B. F. Goodrich Rubber Company). The 8-ply

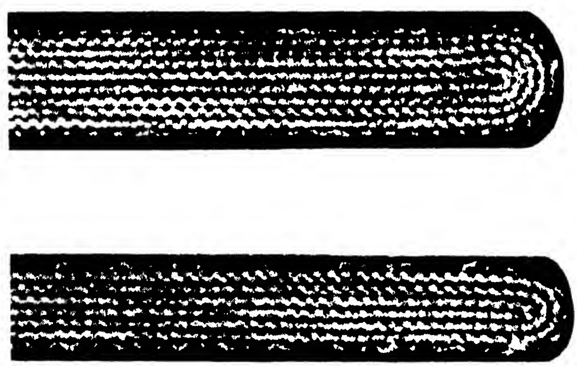


FIG. 20-2. Elevator Belts with Rubber Covers on Both Sides. Covers Cemented and Vulcanized with Tie-gum Construction. (B. F. Goodrich Rubber Company.)

belt has 1/8-inch covers on each side, and the 6-ply belt has 1/16-inch covers. The 6 plies of fabric have been made with skim coats (see page 34), so as to make the belt quite flexible in spite of its thickness and to keep water out of the fabric if the belt should be cut or the covers worn away. The covers are cemented to the body of the belt with the "tie-gum" construction referred to on page 44.

Width of Belts. Belts for wet elevators should be wider than for dry elevating. There are several reasons for this:

1. The water, over and above what fills the voids in the crushed ore, must be provided for in choosing the size of the buckets, and hence the width of the belt.

2. Thin pulps are likely to splash out of the buckets in the pick-up

and be forced out over the lip of the bucket by the resultant of the forces due to centrifugal force and gravity (see page 298). Hence it is safe to use only a fraction of the nominal capacity of the buckets as given in manufacturers' catalogs. Some experienced millmen use only one-third of the nominal capacity.

3. As between a narrow belt with buckets that project far from the belt, or a wide belt with buckets of less projection, the wide belt is preferable; it is less likely to slip on the wet pulley, and the pressures which tend to cut or wear the belt are less per square inch of belt surface.

See page 344 on the pick-up and discharge of mineral pulps.

Belts for Continuous Bucket Elevators. The steel-plate buckets used in these elevators weigh more per foot of belt than buckets in

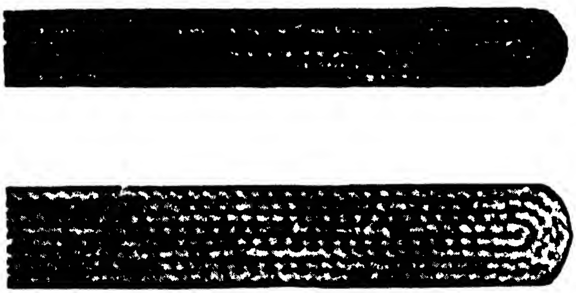


FIG. 20-3. Elevator Belts with Heavy Duck for Hard Service. (B. F. Goodrich Rubber Company.)

centrifugal discharge elevators; the backs are always flat, the projection generally greater than with round-bottomed buckets, and the distance A (Fig. 23-14) relatively smaller compared with the dimensions of the bucket. Consequently the pull on the bolts is severe, and unless the belt is stiff and strong it may be injured on the pulley side by the pressure of the heads of the bucket bolts; or the bolts may pull clear through the belt.

For these reasons, and because of the great wear on the belt surface, belts for heavy continuous bucket elevators should be made of duck heavier than the 32-ounce used in most rubber belts for elevator service. Rubber belts with 36-ounce and 42-ounce duck are made for heavy stone and ore elevators with continuous buckets, generally with thin rubber covers for dry work. Fig. 20-3 shows two specimens; the 6-ply belt has warp threads larger and fewer per inch

than in ordinary elevator belts, and the filler threads are correspondingly closer and thicker. The 8-ply belt shows a very heavy close-woven duck.

Stitched canvas belts made of standard 32-ounce duck—37-ounce on the basis on which rubber belts are graded (see page 60)—show great resistance to the tendency of the bolts to pull through the belt, especially when saturated with a Class 1 drying compound (see page 62) and properly stretched and cured. For strength of canvas belts see Chapter 3.

Balata belts made of 38-ounce duck have a density and strength which fits them for work of this kind. For strength of balata belts see Chapter 3.

Canvas belts are generally preferred for handling cotton seed and for oily materials like spent fuller's earth.

Hot sand from driers requires special heat-resisting belts or insulating pads between buckets and belt.

Before you buy an elevator belt, let the belt maker know what you want to do, and let him give his advice. Your own unassisted judgment may not be good enough.

CHAPTER 21

FASTENING BUCKETS TO BELT

Fastening Buckets to Belt. Since a belt, in passing around a pulley, forms an arc tangent to the flat back of a bucket, theoretically the bolts which fasten on the bucket should be in a single row across the back of the bucket. This is the rule on all elevators handling grain and similar materials: the bodies of grain buckets are made of tin plate or light sheet steel, No. 24 or No. 26 gauge, and the holes are punched in a single row through the fold of metal or reinforcing band at the top of the back. If the entire back were thick, as in malleable-iron buckets or in seamless steel buckets, it would be possible to get a better grip on the belt by having bolts in two rows an inch or two apart, but unless the pulleys are relatively large, or the rows close together, this practice will injure the belt or the bucket.

From Fig. 21-1, representing a belt on a pulley, it is evident that if the holes are punched in the bucket for two rows of bolts, and if the belt while straightened out is punched to match, the holes no longer match when the belt runs over a pulley; the holes in the belt are pulled out of parallel by the bend of the belt and a gap opens between the belt and the back of the bucket. The difference between the chord A and the arc B is not great (see Table 21-A), but for a 7- or 8-ply belt which is ½ inch thick, the difference of the arcs B and C is more than 1/16 inch for a row spacing of 2 inches on pulleys of 34-inch diameter or less. Bolt holes in buckets are made 1/16 inch larger than the bolts and allow some freedom for the bolts; but too much movement of that kind will cut off the bolts or wear out the backs of the buckets where the nuts bear against them.

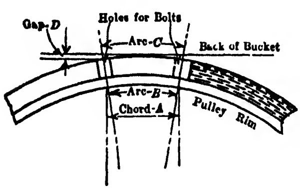


Fig. 21-1. Double Row of Bucket Bolts Passing Over Pulley. (See Table 21-A.)

The gap D is 1/32 inch or more for a row spacing of 2 inches in a belt as it bends over a pulley of 34-inch diameter or less. This dimension is a measure of the tendency of the bolts to bend the backs of the buckets or pull through the belt when passing over a pulley. There is

a definite pull on these bolts when the bucket digs its load out of a boot, and when the material is hard and lumpy and the belt speed high it is not uncommon to find the belt on the pulley side injured by the movement of the bolts; or the bolts may even pull clear through the belt.

Continuous Buckets on Belt. The large steel-plate buckets used on stone elevators are often so heavy that it is necessary to fasten them to

TABLE 21-A

DIMENSIONS OF FIG. 21-1 FOR TWO ROWS OF BOLTS 2 INCHES APART

Diameter of Pulley, inches	Arc *B*—Chord *A*, inches [1]	Arc *C*—Arc *B* For Belt $\frac{1}{2}$ Inch Thick, inches [2]	Gap *D*, inches [1]
12	0.010	0.167	0.091
24	0.0025	0.083	0.044
36	0.001	0.056	0.030
48	0.0004	0.042	0.022
96	0.0002	0.021	0.010

[1] Proportional for spacings other than 2 inches (approximately).
[2] Proportional for thicknesses other than $\frac{1}{2}$ inch.

the belt by two rows of bolts to prevent them from pulling loose or injuring the belt. These rows are midway in the height of the back of the bucket; whether they are 1 or 2 inches apart, the head and foot pulleys should not be less than 30 inches in diameter, and the loading should be arranged so that the buckets receive material direct from a chute while on a straight run and never have to dig out of a boot or even come into contact with material spilled under the foot wheel.

Malleable-iron Buckets on Belts. In belt elevators handling ores and other gritty and heavy materials it is customary to use malleable-iron buckets; and when the projection of the lip from the belt exceeds 5 or 6 inches the pull on the bolts from the digging and from the weight of the bucket is often too much for a single row of bolts. In spite of the disadvantages of two rows of bolts, it is advisable to have two rows for buckets 8 by 5 inches and larger. If the rows are not over 1 inch apart, the difference between *B* and *C* (Fig. 21-1) is about $\frac{1}{32}$ inch for all belts not over $\frac{1}{2}$ inch thick on pulleys at least 24 inches in diameter, and *D* is about $\frac{1}{64}$ inch under the same conditions. For spacing of bolts see Table 21-B.

At any rate it is better to hold the buckets on securely and avoid accident, even though the wear on the belt may be somewhat greater.

It may be said, however, that the tendency to cut the belt at A (Fig. 21-2) may be less when the bucket is held by two rows of bolts. This is particularly true on the down run of belt; heavy buckets upside down tend to swing away from the belt with the leading edge as a pivot (see G, Fig. 21-2). Some bits of material are always picked up in the boot where the bucket gaps away from the belt and carried between the belt and the back of the buckets ahead of the bolts. Any pressure there when the bucket is inverted may force particles into the belt and damage it, unless a pad of belting (see C, Fig. 21-2) is placed back of the bucket or unless the bolts are kept tight.

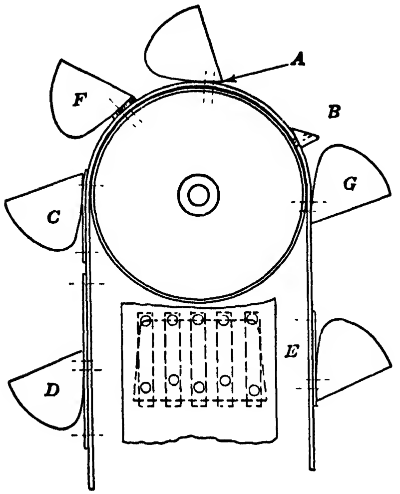


FIG. 21-2. Devices to Protect Elevator Belts from Injury Where Buckets Are Bolted On.

Besides bolts, other methods have been used or proposed to fasten buckets to belts. Oliver Evans (see page 338) used straps and buckles; Griscom, in 1896, patented a sheet-metal clip fastened to the bucket and bent over edges of the belt. Another inventor proposed, in 1914, to fold the belt into a loop at each bucket and fasten the bucket to the loop. Filling and discharge of the bucket were apparently secondary items in this design. None of these schemes is in use at present; others could be mentioned, but they are not so cheap, simple, or handy as the bolt fastening.

Pull on Bolts. When a bucket with a flat back is fastened to a belt by a single row of bolts there is only a short contact between it and the belt as it bends around a pulley. When the strain of digging comes on a bucket there is a pull on the bolts which in Fig. 21-3 is measured by $T = \frac{RP}{A}$, where R is the resistance to the travel of the bucket through the boot. The value of T for a given bucket can be reduced by making the bucket with a curved back as in the Buffalo bucket (Fig. 18-5); then, in Fig. 21-3, $T = \frac{RP}{B}$. Since B may be two or three times A, the pull on the bolts is only one-half or one-third as much, and where the digging is severe, as in grain elevator boots filled up above the

level of the foot shaft, or at the foot of marine legs (Fig. 23-13) digging cargo grain, it is advisable to use buckets with the curved back.

To some extent, the same may be said of buckets fastened on by two rows of bolts. The bucket then has a longer contact with the belt and the pull on the bolts is less.

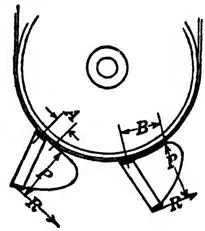


FIG. 21-3. Pull on Bucket Bolts Dependent on Shape of Bucket.

The elevator bucket shown in Fig. 18-17 is used on grain elevators in Europe; it has a flat back, but the sheet forming the bottom is extended and flanged at the lower end to act as a prop against the belt in going around the foot wheel. With this construction, the distance corresponding to *A* or *B* (Fig. 21-3) is even greater than in the Buffalo bucket, and the pull on the bolts is therefore less. At the same time, in going over the head wheel, the extended bottom sheet acts as a deflector for the discharge from the following bucket, and the buckets therefore can be placed close together or slightly overlapping.

Table 21-B (Link-Belt Company) gives manufacturers' standard spacing of holes in sheet-steel and malleable-iron buckets for attaching to belts.

Damage to Belts Where Buckets Are Bolted on. Heavy steel or malleable-iron buckets fastened to belts and used to elevate hard, sharp ores, coke, and minerals often cut the belt at *A* (Fig. 21-2), especially if the bolts get loose and allow the bucket to move with relation to the belt. Bolts loosen from several causes:

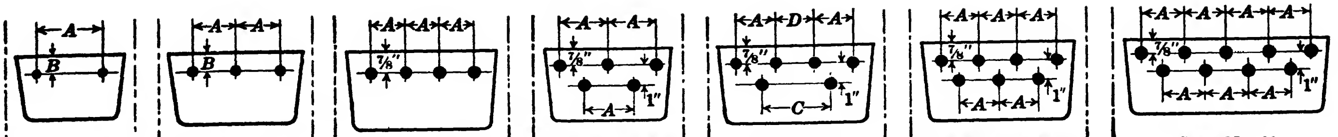
1. Nuts slacking off.
2. Metal of the bucket wearing away under the nuts.
3. Belt wearing thin, or back of bucket wearing thin, from moving on each other.
4. Bolt heads wearing deep into the belt.

When the bolts get loose, centrifugal action at the pulleys tends to throw the bucket outward with the leading edge of the back as a fulcrum, and the pressure there drives hard particles into the belt; or the edge, if sharp, may cut the belt.

There is generally some wear between the belt and the flat back of the buckets used in elevating hard, gritty materials. Some of the discharge gets into the gap which opens there in going over the head wheel, or dribbles into it when the buckets are inverted on the descending run and hang away from the belt (*G*, Fig. 21-2). The same thing happens at the foot wheel when the buckets gap away from the belt.

TABLE 21–B

Punching of Buckets for Belt Elevators
(Manufacturers' Standard)



Styles No. 1 & 7 Styles No. 2 & 8 Style No. 3 Styles No. 4 & 9 Style No. 5 Styles No. 6 & 10 Style No. 11

For all styles, width of belt = bucket length + 1 in. up to 16 in., and bucket length + 2 in. for 16 in. and over. Four-ply belt takes ¾-in. bolt length; 5- and 6-ply take 1 in.; 8-ply takes 1¼ in. Use one leather washer on each bolt between belt and bucket.

Dimensions in Inches

For Salem and Other Light Steel Buckets When Handling Grain or Similar Free-flowing Material

Bucket Length	A	B
Style 1		
2½	⅞	½
3	1⅜	½
3½	1⅞	½
4	2⁵⁄₁₆	⅝
4½	2¾	⅝
5	3³⁄₁₆	⅝
5½	3⅝	⅝
6	4¼	⅝

Bucket Length	A	B
Style 2		
7	2½	¾
8	3	⅞
9	3½	⅞
10	4	⅞
Style 3		
11	3	....
12	3¼	.....
13	3½	...

Bucket Length	A	C	D
Style 4			
14	6	....	....
15	6½	....	.....
16	7	.....	....
17	7½	.....	.
Style 5			
18	6¼	9⅜	3⅛
19	6⅜	9⁹⁄₁₆	3³⁄₁₆
20	7	10⅛	3½

Bucket Length	A	C	D
Style 5 (Continued)			
21	7½	11	3½
22	7¾	11¾	4
Style 6			
24	7	.	.
26	7¾	..	..
28	8¼	.	.
30	9	..	..

Use ¼-in. bolts and ¼-in. leather washers.

For Malleable-iron A, AA, B, and C or Steel Buckets When Handling Coal or Similar Lumpy Material

Bucket Length	A	B
Style 7		
3	1⅜	¾
4	2⁵⁄₁₆	¾
5	3³⁄₁₆	1
6	4⅜	1
Style 8		
7	2½	1

Bucket Length	A
Style 9	
8	3
9	3
10	3½
11	4
12	4½
...	...

Bucket Length	A
Style 10	
13	3½
14	4
15	4
16	4½
17	4½
18	5

Bucket Length	A
Style 11	
19	4
20	4
21	4½
22	4½
23	5
24	5

Use ¼-in. bolts and ¼-in. leather washers up to 10-in. width; ⁵⁄₁₆-in. bolts and ⁵⁄₁₆-in. leather washers for 10-in. widths and up.

These spacings between holes and bolt sizes also correct for steel continuous type buckets but position of holes should be changed to one-half the depth of bucket.

TABLE 21-B (*Continued*)

STANDARD PUNCHING FOR "GRAIN" TYPE BUCKETS MOUNTED ON BELTS

Dimensions in Inches

Length of Bucket	Minneapolis "V"			Buffalo			Rialto			Steel Grain		Ear Corn Without Brace		Ear Corn With Brace		
	Number of Holes	Center to Center of Holes		Number of Holes	Center to Center of Holes		Number of Holes	Center to Center of Holes		Number of Holes	Center to Center of Holes	Number of Holes	Center to Center of Holes	Number of Holes	Center to Center of Holes	
3	2	1⅜														
3½	2	1⅞														
4	2	2 5/16														
4½	2	2¾														
5	2	3 3/16								2	3 1/16					
5½	2	3⅝								2	3¾					
6	2	4⅜								2	4¼					
7	3	2 11/16								2	5					
8	3	3 1/16								3	3					
9	3	3⅝								3	3½					
10	3	4⅛		3	3 15/16		3	3 15/16		3	3 15/16					
11	3	4 11/16								4	3	4	3			
12	4	3⅜		4	3¼		4	3¼		4	3 5/16	4	3 5/16			
13												4	3 9/16			
14	4	4		4	4		4	4		4	3 15/16	4	3 15/16			
15				5	3¼		5	3¼				5	3¼			
		Two Center Holes	All Others		Two Center Holes	All Others		Two Center Holes	All Others						Two Center Holes	All Others
16	6	3	2¾	6	3	2¾	6	3	2¾			5	3⅜	6	3	2¾
18	6	4⅜	2⅞	6	4⅜	2⅞	6	4⅜	2⅞			6	3 9/16	6	4⅜	2⅞
19												6	3 13/16			
20	6	4½	3⅜	6	4½	3 5/16	6	4½	3 5/16			6	3 13/16	6	4½	3 5/16
22				6	4 11/16	3¾	6	4 11/16	3¾							

Unless otherwise specified all buckets will be punched as per above dimensions. If buckets are ordered for replacements, check spacing of present punching and specify centers, to avoid repunching of belt.

All punching is for ¼-in. bolts. Refer to Fig. 21–10 for types of bolts.

Protective Devices. Various devices are in use to protect the belt from the wear mentioned above. A rubber cover on the belt is often the cure and may pay for itself in the longer life of the belt and the lower cost of repairs and renewals. In one elevator where the life of the belts averaged only a few months, triangular prisms of wood (*B*, Fig. 21-2), equal in length to the width of the belt, were fastened on by wood screws below each bucket to prevent small bits from getting between the belt and the bucket. This scheme costs only a few dollars, but it more than doubled the life of the belt. For protective strips for continuous buckets, see page 336.

In the western mining country where centrifugal discharge belt elevators are extensively used for handling not only fine material, wet or dry, but also lump ore up to 3-inch size, it is common practice to place a pad of old belt between the bucket and the elevator belt (*C*, Fig. 21-2). Sometimes the bucket bolts do not go through the elevator belt, but only through the pad, which is then extended above and below each bucket to protect the belt from abrasion (*D*, Fig. 21-2). This is open to the objection that, unless the bolts are kept tight, fine stuff will work in between the pad and the belt and, being confined there, will rub and injure the belt.

In wet elevators a narrow strip of old belt is used with each bucket bolt (*E*, Fig. 21-2) to space the bucket away from the belt and leave room for water to wash away the grit from behind the bucket. Soft rubber washers (*F*, Fig. 21-2) have been used for the same purpose and to keep the leading edge of the back from digging into the belt. Buckets mounted on thick washers cannot do heavy digging in a boot.

Troubles from Buckets Working Loose. Besides the cutting of the belt mentioned above, there are other troubles, still more serious, due to nuts working loose and coming off the bolts. When only a few bolts in a bucket stay tight, the strain on them is excessive and they may damage the pulley side of the belt or pull through the belt, or break off. A bucket falling off into the boot may rip other buckets off and cause a breakdown; buckets thrown off into the head chute may damage machinery fed by the elevator, or cause spouts and hopper gates to choke. In grain elevators it is good practice to put a grating near the top of the head chute to catch such loose buckets, as well as the sticks of wood, pieces of paper, etc., which often pass from grain cars into the elevator leg. A clean-out door should be provided to give access to the screen for examination and cleaning (see Fig. 23-25).

Inspection of Buckets. Trouble and expense can be avoided by inspecting buckets and their fastenings regularly and tightening or replacing the loose or broken bolts. If the belt shows signs of being

cut or worn by the buckets or bolts new spots can be made to take the wear by shifting all the buckets to new positions somewhere between the old ones. Some men who install belt elevators punch the belt at the start for two or three settings of buckets; that is, if the buckets are to be spaced 18 inches apart, the new belt would be punched for bolts every 9 or 6 inches. This saves time and money when it becomes necessary to shift the buckets.

Width of Bucket and Width of Belt. The usual crown of belt pulleys is $\frac{1}{8}$ inch on the diameter per foot of face—that is, the face of a pulley for a 12-inch belt is $\frac{1}{16}$ inch higher in the center than at the edges. When a belt runs over a crown-face pulley its center is stretched more than its edges, and a bucket fastened to the belt must either bend in the back or tend to pull the bolts through the belt; or else the belt does not conform exactly to the crown along the line of the bolts. In grain elevators the belts are relatively stiff and the buckets light, and the back of the bucket springs slightly to match the crown when the face is not too wide. Sheet-steel buckets of heavier metal do not spring so easily, and malleable-iron buckets are too stiff to spring at all. Hence it happens that when wide stiff buckets are fastened to a belt of the same width the bolts are apt to dig into and injure the pulley side of the belt. Instead of using malleable-iron or stiff steel buckets, say 20 inches or wider on a belt of that width, it is better to use smaller buckets of half the width in double row to give the required capacity. The spacing between consecutive buckets in each row should, of course, be what will give a clean discharge at the head, and if the widths of the buckets in the two rows do not overlap along the center line of the belt there will be no interference with the discharge when the buckets are set staggered.

Buckets in Double Row. In grain elevators and other elevators having wide belts it is general practice to use double rows of staggered buckets when belts are wider than 22 inches (see Fig. 21-4). The advantages of this construction are as follows: (1) The buckets avoid the crown of the pulleys. (2) The buckets are stronger and stiffer and the fronts are less likely to pull out under heavy load, or on striking an obstacle like a stick of wood in the boot. (3) If a bucket is spoiled by any mishap, the cost of replacing it is less than if the accident happened to a bucket of double the width. (4) The work of pulling the buckets through a deep mass of material in the boot is less than for wide buckets in single row.

Belts in Casings. When elevator belts run in casings it is usual to make the width of the belt 2 inches more than the width of the bucket, or 1 inch more if the bucket is not over 6 inches wide; then

there is a margin of belt on each side which keeps the bucket from striking the casing. Of course, casings are supposed to be made with clearance enough to avoid interference, but if they should twist or get out of line, it is better that the belt should rub than that the bucket should strike.

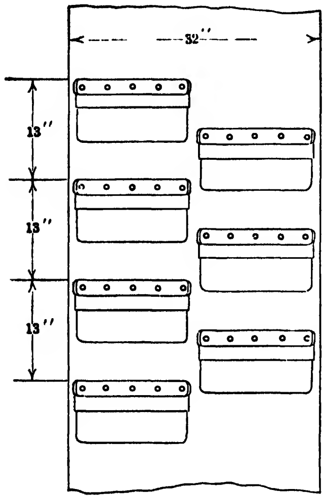


FIG. 21-4. 15-inch Buckets Spaced 13 Inches in Double Row on 32-inch Belt.

If there is no casing, or if the elevator is enclosed in a roomy housing, the margin of belt is not necessary as a guard; but where the head pulley gets wet or works in a cloud of dust, a belt no wider than the buckets is more likely to slip than a belt made a few inches wider. The coefficient of belt contact is less under such circumstances than when the head pulley is dry and free from dust.

Buckets to Match Crown of Pulleys. A patent was granted in 1916 (No. 1,194,308) on a bucket with its back curved to match the crown of the pulley. Such buckets are not in practical use. The idea might be applied, with some advantage, to wide buckets, but, for reasons stated above, it is better to use narrower buckets in two rows instead of very wide buckets in a single row. Aside from that, there are mechanical difficulties in making sheet-steel buckets with the curve for crowning combined with the curve in the bottom of the bucket. It would be still more difficult to apply the idea to grain buckets with a curved back like Fig. 18-5, but malleable-iron buckets could be made with that feature if there were a demand for it.

Joining Ends of Elevator Belts. The requirements for a good joint for elevator belts differ in some ways from those for conveyor belts. In the latter, the fastening must be flat on both sides because both sides of the belt run over the idlers. In elevator belts, the fastening need not be flush or even flat on the bucket side of the belt; the pull in the belt per inch per ply is often greater than in conveyor belts, and a stronger splice is needed. Another difference is that the take-up in an elevator is usually shorter, resplicing must be done oftener, and a fastening that can be taken apart and put together readily in a confined space is preferable. For these various reasons, elevator belts are joined with bolted splices rather than with clinch hooks or clinch rivets.

One of the oldest and simplest fastenings is shown in Fig. 21-5. It is

very strong and resists shocks, but it is not easy to apply to belts over 5-ply, and when it is put on thick belts, the bending in going over pulleys is localized at the corners of the flat bars, even though they are rounded. This leads to breaking the warp threads and to cracking the belt crosswise. This joint should not be used on canvas belts; they are too stiff and the duck is too heavy to stand the bending.

Lap Joints. The commonest joint on grain elevator belts and for heavy continuous bucket elevators is a plain lap splice (Fig. 21-6). The lap may cover a distance of 4 feet or more and is held together by the bucket bolts. The joint is simple and strong and requires no extra parts; when the belt becomes too long, it is shortened by one or two bucket spacings, bolted together again, and the excess length is

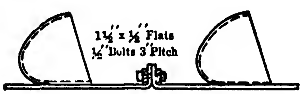


FIG. 21-5. Bolted Clamp Joint for Elevator Belt.

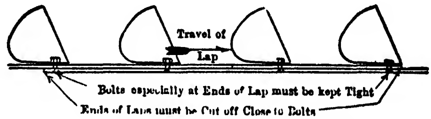


FIG. 21-6. Bolted Lap Joint.

cut off. It is successfully used on 6- or 7-ply grain elevator belts, but on 8-ply belts, the double thickness at the lap measures about 1 inch, and when this bends over pulleys, the two thicknesses tend to move on each other and there are strains which work the bolts loose in the holes and cause wear on the belt under the bolt heads and nuts. If the laps are not held tight together, material gets between them, the bolts pull through the belt, and the belt may break across the line of bolt holes. To prevent this, the bolts must be kept tight, especially those at the ends of the laps.

In stone elevators, sand and grit, getting between the laps, has been known to grind the belt partly through until it broke. The same thing has happened when an end of belt at the lap was not cut off close to the bolts; bits of stone wedged behind the projecting flap of belt and finally cut the belt so badly that it broke. In this particular case the belt was 12-ply and was lapped the wrong way for the travel of the belt. The right way is shown in Fig. 21-6; the cut end of belt on the inside of the lap should trail over the pulleys, not run against them.

Butt-strap Joints. The ends of heavy belts are frequently butted and the joint made by a piece of belting as wide as the elevator belt and two or three times as long as it is wide. In elevators with spaced buckets the length of the butt-strap may be twice its width and it is riveted on. In continuous bucket elevators it usually extends under two buckets on each side of the butt and is secured by the bucket bolts

(Fig. 21-7). This joint has the same merits and the same defects as the lapped joint mentioned above; in addition, a bucket may have to be left off at the joint to make room for the double row of bolts where the ends of the belt come together. These bolts must be close to the cut ends of the belt to prevent material from working in between the belt and the butt-strap, and they must be kept tight.

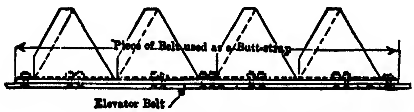


FIG. 21-7. Butt-strap Joint.

The Jackson belt fastener (Fig. 21-8) consists of a series of stamped steel plates each with two countersunk head bolts, two oval cup washers with prongs, and two sleeve nuts. The ends of the belt are cut square with a thin piece of canvas belt as a templet; the bolt holes are punched in them. The bolts, with the oval washers on them, are inserted from the pulley side of the belt; then, on the bucket side, the canvas templet is laid for a cushion and the plates are put on. When the nuts are screwed tight, the cup washers pull the belt up into the concaves of the top plates, the sleeve part of the nut wedges the warp threads together, and the belt is held tight. The shape of the top plates is such that the two bolts do not stand parallel to each other, but at an angle to make them approximately radial in passing around pulleys of a size suited to the fastener. This prevents the bolts from moving in the belt or tearing it in passing over pulleys.


FIG. 21-8. Elevator Belt Joined with Jackson Fasteners.

The Jackson joint is made for all thicknesses of belts; it is strong, and when applied to stiff thick belts, it is not likely to injure them as sometimes happens with lap joints or butt-strap joints. It is one of the best joints for stitched canvas or balata elevator belts.

Fig. 21-9 shows a butt-strap joint which has been used on belts handling very gritty material. The ends are joined by a metal fastener of some kind, and a strip of belt, usually lighter than the elevator belt, is placed over the joint and under the buckets to act as a cushion for the buckets, to protect the metal fastener from the grit, and also to strengthen the joint.

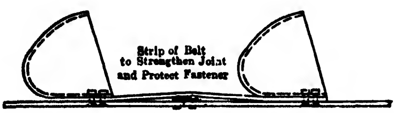


FIG. 21-9. Elevator Belt with Bolted Fastener and Butt-strap.

Modified Jackson Joint. A splice used by a western mining company for heavy elevator belts is a butt joint covered on the bucket side by two curved cast-brass plates $\frac{9}{16}$ inch thick, $11\frac{1}{2}$ inches long along the belt, and each of about half the width of the belt in the other dimension. There is a small space between the two plates over the crown of the pulley. The plates are curved to 30-inch radius for use with 60-inch pulleys and are made with a reverse curve at the edges to avoid cutting the belt. They are fastened on by 4 rows of $\frac{1}{2}$-inch diameter countersunk head bolts with oval cup washers of the Jackson type on the pulley side of the belt.

FIG. 21–10. Flat-head Elevator Bucket Bolts.

Elevator Bucket Bolts. Fig. 21-10 shows the flat-head bolts used for fastening buckets to belt. For 4-ply belt use $\frac{3}{4}$-inch; for 5-ply or 6-ply, use 1-inch length; for 8-ply, use $1\frac{1}{4}$-inch length.

Leather Washers. Leather washers are used on elevator bolts, between the bucket and belt, to act as an elastic cushion when buckets are passing over the pulley, and provide open spaces which prevent fine material from accumulating or wedging between belt and bucket. Fitting the bolts tightly prevents moisture working into the body of the belt at the bolt hole. Putting rubber cement in the bolt holes is a help.

CHAPTER 22

DRIVING BELT ELEVATORS

Drive by Head Pulleys. In the design of a belt and bucket elevator the diameter of the head wheel must always be considered in connection with its speed; the two are not independent, but for a good discharge of materials, they hold rather definite relations to each other. These relations, for the usual working conditions, are given in tables in Chapter 17, and the conditions for which it is proper to use these tables are described in Chapters 17 and 19.

The theory of belt driving is based on the assumption that so long as the belt bends freely to the curvature of the pulley the driving effect is independent of the diameter of the pulley. This is confirmed in practice by Haddock's experiments referred to in Chapter 5. If the diameter of the pulley is at least four or five times the number of plies in the belt, the tractive effect does not vary with the size of the pulley; hence the general rule is that a 6-ply belt, for example, should have a head pulley at least 24 inches in diameter, better still 30 inches, and so far as the internal wear in the belt is concerned, the larger the diameter the better. It is a fact, however, that most elevator belts fail for reasons other than internal wear, and there are disadvantages in making the head pulleys too large; the head group takes up too much space, the head chute drops lower, the casing is larger, supports become heavier, larger driving machinery is needed, and for most of these items the cost is greater.

The driving contact between a head pulley and an elevator belt depends upon the angle of wrap and the coefficient of friction between the belt and the rim of the pulley. Usually the angle of wrap is 180°; hence the general expression for the ratio of tensions on the up side T_1 and the down side T_2 (see page 145) becomes, for a coefficient of friction of 0.25, $\frac{T_1}{T_2} = 2.19$, and for a coefficient of 0.35, $\frac{T_1}{T_2} = 3.00$ (see Chapter 5).

That is, a head pulley will drive an elevator belt when the pull on the down side is from one-half to one-third the pull on the up side, assuming that pulley and belt are clean and dry. If the pull on the down side is more than one-half the pull on the up side, the drive is more certain to act under unfavorable conditions, such as dust, dirt on the belt, wet pulley face, or pulley side of belt rough and torn.

Coefficients of Friction. The coefficients 0.25 and 0.35, mentioned above, are satisfactory in calculations for belt conveyors and generally give good results in calculations for belt elevators also. The general formula for the value of $\frac{T_1}{T_2}$ assumes that f, the coefficient of friction, depends solely on the nature of the surfaces in contact. Experiments by Wilfred Lewis (*Transactions A.S.M.E.*, Vol. 7) show that, as the load increases and $T_1 - T_2$ becomes greater, the coefficient of friction also increases, although the belt creep and the belt slip increase at the same time. Table 22-A abstracted from Carl Barth's comment on the

TABLE 22-A

VARIATION OF COEFFICIENT OF BELT FRICTION WITH SLIP AND CREEP ON DRIVING PULLEY

1 Experiment Number	2 Tension per Square Inch Belt at Rest, pounds	3 Tension per Square Inch, on Tight Side, T_1 = pounds	4 Tension per Square Inch, on Slack Side, T_2 = pounds	5 Observed Loss in Slip and Creep, per cent of belt travel	6 Calculated Loss Due to Creep Alone, per cent of belt travel	7 Coefficient of Belt Friction Corrected for Centrifugal Force
60	81 6	125 33	58.67	0 5	0.41	0.25
61	81 6	131 42	46 58	0 9	0 53	0.34
62	81 6	142 00	42 00	1 7	0.62	0.41
63	81 6	152.41	35.75	3 0	0.73	0.49
65	81 6	179 92	29 92	12 0	0 91	0.61
66	127.5	177.42	77 42	0.5	0.52	0.27
68	127.5	198 25	64 92	0 8	0.69	0.37
69	127 5	208 77	58.67	1.0	0.77	0.42
70	127.5	219.08	50.75	1.7	0.87	0.47
71	127 5	229 50	46 17	2 6	0.95	0.54
72	127 5	244.08	44.08	3 8	1 02	0 57
73	127.5	256.58	39 92	3.5	1.10	0.62
74	127.5	252.42	35.75	8.6	1.13	0.68
75	127.5	283.66	33.67	15.2	1.25	0.72
128	343.5	511.3	227.0	0.5	0.85	0.26
131	343.5	557.0	187.2	1.1	0.99	0.35
133	343.5	589.5	162.4	1.8	1.30	0.41
134	343.5	603.0	148.2	2.7	1.39	0.45
135	343.5	618.0	134.0	5.1	1.49	0.49

NOTE: Since 1-ply thickness in a rubber or canvas belt is about $\frac{1}{16}$ inch, divide the figures in columns 2 and 3 by 16, to estimate the equivalent tension per inch per ply in the belts, had they been of fabric instead of leather.

Lewis experiments (*Transactions A.S.M.E.*, 1909) gives in column 7 values of f for leather belts on clean iron pulleys at 800 feet per minute; they vary from 0.25 to 0.72.

These experiments were made with clean leather belts; but, reasoning from the known behavior of various kinds of belt in the transmission of power, there is no reason to think that rubber or other fabric belts would act differently, except for possible differences in the numerical values of the coefficient of friction. It is probable that the coefficients for these belts on clean iron pulleys or on pulleys covered or lagged with rubber are larger than 0.25 and 0.35, respectively, but when the belt is wet, as in elevators that handle wet ores, or when it becomes covered with granules of dirt, or when the interior of the casing is thick with dust as in grain elevators, then the coefficients are less than for clean pulleys. How much less we do not know. Haddock, in 1908, made experiments (*Transactions A.S.M.E.*, Vol. 30) with a 12-inch 4-ply belt wrapped 180° on plain and rubber-covered pulleys dusted or coated with various substances; but his values (see Table 22-B) are quite erratic and can only be taken to show that the coefficients are less when the pulley rim is dusty or damp than when it is clean.

For dusty work the coefficients of friction for rubber or fabric belts

TABLE 22-B

TRACTIVE EFFECT EXPRESSED AS PERCENTAGES BASED ON CLEAN DRY SURFACES OF RUBBER BELT AND IRON PULLEYS

(Haddock's experiments, 1908—)

Condition of Contact Surfaces	Rubber Belt on Iron Pulley, per cent	Rubber Belt on Rubber Lagged Pulley, per cent
Clean, dry	100	108
Clean, damp	92	101
Covered with dry coal dust	92	80
Covered with damp coal dust	76	52
Covered with dry clay	63	55
Covered with damp clay	55	65
Covered with dry slate dust	68	138
Covered with damp slate dust	85	147
Covered with dry, sharp sand	42	80
Covered with damp, sharp sand	62	80

on iron pulleys may be taken at 0.20, and for rubber-covered pulleys 0.27. For these values the corresponding ratios of $\frac{T_1}{T_2}$ are 1.87 and 2.33, respectively. For wet work, the coefficient may be called 0.20, whether the pulley is bare or lagged.

It is certain that a pulley covered with rubber belt will pull more than a plain iron pulley if the work is dry, and although some engineers doubt the value of lagging on head pulleys of wet elevators, there are some practical advantages in it, even though it may not increase the value of f, the coefficient of belt contact. When belts wear down to the fabric on the pulley side and become rough, and when bolt heads project beyond the belt surface, then the contact with an iron face is not continuous, but interrupted, and the belt if wet is more likely to slip; but if the pulley is covered, the rim accommodates itself better to these irregularities and the driving contact is better. This is especially true if the lagging is not ordinary friction-surface rubber belt, but standard belt lagging which is several plies of fabric covered with a layer of rubber.

The application of the above to the design of an elevator can be discussed best by reference to an example.

Pull at Head of Grain Elevator. The unbalanced torsional pull at the rim of the head pulley, which measures the power required to drive the elevator belt, is the sum of several items:

1. Weight of grain in the lifting buckets = G.
2. Drag of buckets through the grain in the boot = B.
3. Friction of foot shaft and pulleys = F.

The total pull in the elevator belt is the torsional pull plus the weight of belt and buckets on the rising side.

It is easy to calculate item G directly, but not B and F. F is small and may be estimated, but B must be derived indirectly from power tests of the elevator.

Calculation of Pull from Power Readings. A certain grain elevator had a nominal capacity of 12,000 bushels per hour; the calculated capacity based on all buckets lifting their full load and discharging it without spill was 14,560 bushels per hour. When the following test was made, the elevator was lifting wheat at its regular rate; assuming a possible loss of 10 per cent in filling the buckets and in spill at the head, the work was probably at the rate of about 13,000 bushels per hour. The lift was 216 feet.

Power delivered to the motor		80 kw.	
80 kw. $= \dfrac{80 \times 1000}{746} =$		107.2 hp.	
Motor loss (efficiency about 94 per cent)	= 6.4 hp.		
Estimated loss in silent chain drive, countershaft, and rope drive	= 5 4 hp.		
Estimated loss in turning head shaft at 700 f.p.m. belt speed	= 2.4 hp.		
Total losses up to elevator belt		14.2 hp.	
Power delivered to elevator belt			93 hp.
Horsepower to lift 13,000 bushels wheat at 60 lb. per bushel to height of 216 ft.		85 hp.	
Estimated loss at foot shaft		1 hp.	
Horsepower due to load and foot shaft			86 hp.
Horsepower to drag buckets through grain in boot $= B =$			7 hp.

Torsional pull delivered to elevator belt $= \dfrac{93 \text{ hp.} \times 33{,}000}{700}$ = 4400 lb.

Weight of belt on each side	$= 216 \times 6.07 = 1310$ lb.	2700 lb.
Weight of empty buckets each side	$= 345 \times 4 \quad = 1380$ lb.	
T_1 = total tension in up belt (no take-up tension)		= 7100 lb.
T_2 = total tension in down belt (no take-up tension)		= 2700 lb.

In the example above, if we assume $f = 0.25$ and $\dfrac{T_1}{T_2} = 2.19$, then for a tension T_2 in the down belt of 2700 pounds, the head pulley would exert a pull T_1 of $2700 \times 2.19 = 5900$ pounds in the up belt. This is far short of the actual pull of 7100 pounds and it is, therefore, probable that the elevator could not have been driven with a bare iron head pulley unless the boot shaft were loaded or screwed down to put an extra tension in the belt. If the boot shaft were loaded with 2500 pounds, 1250 pounds added to T_1 and T_2 would make them, respectively, 8350 and 3950 pounds; then, $\dfrac{T_1}{T_2} = 2.12$ and the plain iron pulley might have driven the belt.

The head pulley of this elevator was actually covered with rubber lagging; assuming $f = 0.35$, then if $T = 2700$ pounds, $T_1 = 2700 \times 3.00 = 8100$ pounds. This is an excess of 1000 pounds over the actual pull in the belt or a margin of $\dfrac{1000}{4400} = 22$ per cent above the work of digging and elevating the grain.

Since the foot of this elevator was loaded with about 1000 pounds by a weighted take-up the normal values of T_1 and T_2 in operation were probably about 7600 pounds and 3200 pounds, respectively. If we say that $f = 0.35$, and the ratio of tensions corresponding to that value is 3.00, then the pulley would exert a belt pull of $3200 \times 3.00 = 9600$ pounds, or an excess of 2000 pounds over the normal value of T_1 for such contingencies as overload in the boot, loss of driving contact due to dust in the casing, etc. If f should fall to 0.30, with 3200 pounds on the down side, the pulley would drive the belt with a force of 8200 pounds, and there would be a margin of 600 pounds above the normal value of T_1; if f fell to 0.27, the pulley would exert a pull of 7600 pounds in the up belt, just enough to drive it.

Value of Take-up Tension. When a grain elevator 200 feet high, capacity 12,000 bushels per hour, is fitted with an automatic weighted take-up boot, the boot shaft may move vertically 2 or 3 inches during operation between no load and full load with a good new belt. This shows that the elastic stretch may be 1 or 2 inches per hundred feet in such service, and it also indicates that, in an elevator equipped with take-up screws, it is not possible to maintain a fixed minimum tension T_2 in the down belt, when part of that load is added, as is usual, at the foot. But when T_2 is kept constant by a weighted or loaded take-up, dust in the casing or the condition of the belt or of the lagging on the pulley rim may cause f to vary within rather wide limits without affecting the certainty of the drive. If T_2 decreases, T_1 diminishes also, but in greater amount, because the working ratio $\frac{T_1}{T_2}$ is between 2 and 3. When T_1 diminishes, the digging power of the elevator or its lifting capacity falls off, and it may choke. Hence, in hard-worked elevators, and especially in those which are wet or dusty, it is important to maintain the take-up tension, preferably by weighting the boot shaft.

Calculations of Belt Tensions and Horsepower. The various items which enter into these calculations have been set down in Table 22-C; the following comment will explain what they are and how they should be used.

Item 1. Pull due to Weight of Material. This equals

$$\frac{\text{Pounds of material in 1 bucket} \times \text{Height of elevator (feet)}}{\text{Spacing of buckets (feet)}}$$

or, what comes to the same thing, it equals

$$\frac{\text{Capacity of elevator (pounds per minute)} \times \text{Height of elevator (feet)}}{\text{Belt speed (feet per minute)}}$$

TABLE 22-C
CALCULATION OF ELEVATOR BELT STRESSES AND HORSEPOWERS

Item	Stresses in Up Belt at Head Pulley	Stresses in Down Belt at Head Pulley	Horsepower Required
Lifting the load	(1) Weight of material in buckets	(2) Zero	(3) $\frac{\text{Item 1} \times \text{belt speed,}}{33{,}000}$ or $\frac{\text{Pounds per minute} \times \text{height (feet)}}{33{,}000}$
Weight of empty belt and buckets	(4) Add for this	(5) Add for this	(6) Affects Item (15)
Digging the load or filling buckets	(7) Add as a percentage of height of lift. (See page 367.)	(8) Zero	(9) Add as a percentage of height of lift. (See page 367.)
Friction losses at foot shaft	(10) Add a small percentage	(11) Zero	(12) Add a small percentage
Friction losses at head shaft and in power transmission	(13) Zero	(14) Zero	(15) Estimate and add
Air resistance	(16) Generally very small	(17) Zero	(18) Add a small percentage for high-speed elevators only
If belt operates with little or no take-up tension, total pull in belt equals	(19) Sum of items 1 + 4 + 7 + 10 + 16	(20) Item 5	(21) Total horsepower = sum of items 3 + 9 + 12 + 15 + 18
If added tension is needed to make head-pulley drive	(22) Add enough to make proper ratio $\frac{T_1}{T_2}$	(23) Same as item 22	(24) Affects 12 and 15 only as friction losses
If belt operates under added tension, total pull in belt equals	(25) Sum of items 1+4+7+10+16+22	(26) Sum of items 5+23	(27) Same as item 21 plus a small per cent for greater friction losses

Item 3. Horsepower to Lift the Load. These two expressions come from multiplying item 1 by

$$\frac{\text{Belt speed (feet per minute)}}{33{,}000}$$

to convert pounds of pull to horsepower.

Item 4. Pull Due to Weight of Belt and Buckets. The weight of ordinary elevator belt in pounds per linear foot is approximately width (inches) $\times$ number of plies $\times$ 0.03. For accurate weights, see Chapter 3. Weights of malleable-iron buckets are given in Chapter 18. Weights of other buckets can be taken from manufacturers' catalogues.

Item 7. Pull Due to Pick-up. This can be determined only by experience or experiment. An empirical rule based on long practice is that the pull required to pick up coal, ashes, ores, stone, and similar coarse materials from a take-up boot at the speeds of Table 17-B is about equal to the weight of the material carried on $12D$ feet of belt, where D is the diameter of the foot wheel in *feet*. For example, if an ore elevator has a 2-foot diameter foot wheel, the work of pick-up is equivalent to adding 24 feet to the height of the elevator.

Experiments by Hanffstengel (*Förderung von Massengütern,* Vol. 1, 1915) with a short elevator inclined at 30° from the vertical, having a fixed bearing foot with a semicircular bottom and a chute entering at the level of the foot shaft, gave the results shown in Table 22-D. The work of pick-up, stated as foot-pounds per pound of material elevated, is equivalent to, and may be expressed as, feet added to the height of the elevator, because the work (foot-pounds) of elevating the material is the work of lifting the same weight of material through the height of the elevator. The figures of the table are lower than those given by the empirical rule stated above, but it is probable that the rule is better suited to ordinary practice than the results derived from the laboratory test.

Table 22-D shows that with lumpy material the power required for the pick-up was less when the clearance between the bucket and the boot bottom was small, but where the material was fine, like soft coal slack, the small clearance was of no advantage. This points to what is already known from practical experience, that is, that power is saved in elevators for lumpy material if the bottom clearance is less than the least dimension of the pieces handled, so that they cannot wedge under the bucket. On the other hand, however, small clearance cannot be maintained in a take-up boot, and it is often inconvenient to use a fixed bearing boot or a special boot. Small clearance leads to greater wear on the bottom sheet of the boot and the risk that the sheet may

be cut if the buckets are knocked out of shape or if the foot shaft settles from wear in the bearings. For these reasons, it is better to use the empirical rule in estimating the work of pick-up.

TABLE 22–D

WORK OF PICK-UP IN ELEVATOR BOOT

(Adapted from Hanffstengel)

Material Elevated	Bottom Clearance in Boot, inches	Work of Pick-up in Foot-Pounds per Pound of Material	
		120 Buckets per minute	60 Buckets per minute
Soft coal slack...	$2\frac{3}{4}$	4	4.6
Soft coal slack.	$\frac{3}{4}$	more than 4	more than 4.6
Boiler house coal (crushed)	$2\frac{3}{4}$	4.2	8.2
Boiler house coal.	$\frac{1}{4}$	3.3	5.0
Coke (size not stated)	$2\frac{3}{4}$	5.0	10.0
Coke (size not stated)	$\frac{1}{4}$	3 0	5.0

In high-speed grain elevators with foot pulleys one-third or one-fourth the size of the head wheel the grain piles up on the lifting side and the pull on the belt due to pick-up may be equivalent to $10D$ expressed in feet of height, where D is the diameter of the foot pulley in feet. That is, if such an elevator has a 72-inch head pulley and a 24-inch foot pulley, the pick-up is equivalent to adding 20 feet to the elevator.

With relatively larger foot pulleys and speeds according to Table 17-B the work of pick-up for fine dry free-flowing material may be taken as $6D$.

In inclined elevators with spaced buckets run at the speeds of Table 17-C the pick-up is easier and the pull may be taken at $4D$ instead of $6D$.

In continuous bucket elevators run at speeds not over 150 feet per minute and properly loaded from a chute the shock or impact from loading is not great and the added pull in the belt due to it may be taken at $2D$ or $3D$. But if the feed is poor and if the spilled material is allowed to accumulate under the foot wheel the pull may be $6D$ or even more. The same is true if the buckets at the loading point are confined within steel plates forming a stationary "loading leg," in

which event material catching between the buckets and the plates adds to the pull on the belt.

Item 10. Pull Due to Friction Loss at Foot. One or 2 per cent of the total calculated pull in the elevator belt should be enough to cover this.

Item 15. Pull Due to Power Transmission Loss. Allow 5 per cent for each speed reduction from the source of power through belts, chains, or cut gears, and 10 per cent for each reduction through cast gearing. See also page 143.

Item 18. Allowance for Air Resistance. Five per cent should cover this in high-speed grain elevators.

Item 19. Proper Ratios of Belt Tensions. If the elevator has a belt wrap of 180° on a plain iron pulley the head pulley will drive, without tightening the belt, if T_1, which is item 19, is not more than 2.19 times T_2, which is item 20. If the pulley is rubber-covered, then item 19 can be 3.00 times item 20 without excessive slip (see Table 5-K, page 145).

Item 21. Total Horsepower. This is also equal to

$$\frac{(\text{Item 19}-\text{Item 20})}{33{,}000} \times \text{Belt speed (feet per minute)}$$

plus the allowances added in item 15.

Item 22. Calculation of Added Take-up Tension. Suppose that T_1 = item 19 = 2000 pounds and T_2 = item 20 = 800 pounds, then the ratio is $\frac{2000}{800} = 2.5$; it is greater than 2.19, and a plain iron pulley will not drive the belt unless extra tension is added. To find what tension x must be added to each side of the belt to make $T_1 = 2.19T_2$, we say $2000 + x = 2.19\ (800 + x)$, from which $x = 210$ pounds, which is item 22.

Item 23. Added Belt Tension Due to Take-up. The total tension to be added by the take-up is 420 pounds, divided between the up belt and the down belt.

Maximum Belt Tension. The total pull for which the belt should be selected is either item 19 or item 25.

If we assume a unit tension p, then for a belt of width W, the number of plies $= \dfrac{\text{Maximum tension (item 19 or 25)}}{pW}$

Working Tensions for Belts. What is said on page 58 about unit stresses applies generally to elevator belts as well as conveyor belts. Usually the stress can be kept below 25 pounds per inch per ply for 32-ounce duck; but there are many successful elevators in which the

unit stress is 30 pounds. It should not exceed 35 pounds for 32-ounce duck, nor 40 pounds for 36-ounce duck. See Table 6-B on page 169.

Thickness of Belts as Determined by Wear. Elevator belts for coarse materials are usually made some plies thicker than is necessary to transmit the maximum tension, because in general the life of the belt is determined by the external wear and damage it receives (see page 337). An extra thickness acts then as a protection against belt failure by reason of loss of tensile strength; it also gives the belt strength and stiffness to back up the bucket and prevent the bucket bolts from pulling through under severe strain. This is shown by some of the elevators listed on page 327. In elevator 1 the daily tonnage would, if evenly spread over every minute of the 24 hours, require each bucket to be loaded only one-fourth full. If it is assumed that the buckets are at times three-fourths full, the maximum tension corresponding to item 25, Table 22-C, is 7500 pounds, and if the belt is stressed to 26 pounds per inch per ply, 8 plies are required. Actually 11 are used. In elevator 2 the maximum tension is 8100 pounds, requiring less than 9 plies, but the belt has 12. Similarly in elevator 4, the maximum tension corresponding to a load three times the average, based on tonnage, plus the water carried, is 6800 pounds; this requires less than 9 plies, but the belt has 12 plies.

Minimum Number of Plies in Belt. Practical experience has shown that to give good service belts should have a certain minimum number of plies, based on the considerations stated above, regardless of the tensile strength required. Table 22-E is a fair statement of modern practice.

Belt Slip. When a belt slips on the head pulley there is a reduction of elevating capacity and the danger of a choked boot if the feed continues at the normal rate. In this case the belt may slow down, pull some buckets off, and eventually stop. Even though slip may not come to the point at which the belt stops, wear on the pulley side of the belt is sure to follow. In a rubber belt the duck on one side may be frayed or worn off. In a stitched canvas belt the same thing happens or the rubbing glazes the surface on the pulley side of the belt if it has been painted, and if it has been impregnated with a class 1 compound (see page 62) the heat generated by the rubbing is likely to dry up the compound and make the belt brittle. This is not true of class 2 compounds, because they resist the action of heat much better.

The rubber covering of head pulleys is subject to the same wear as the belt, but since the lagging exposes less surface to it than the belt, the damage is likely to be greater. The heads of the bucket bolts often

tear the lagging when slip occurs, and with that is combined the wear due to belt creep (see page 372).

TABLE 22-E

MINIMUM NUMBER OF PLIES IN ELEVATOR BELTS

Adapted from Goodyear

Material Elevator (Spaced Buckets)	Projection, inches			Weight of Duck
	3–4	5–6	7–8	
Free-flowing materials under 50 lb. per cu. ft.	4	5	6	28 oz. or 32 oz.
Coal, earth, sand, etc., under 75 lb. per cu. ft.:				
1 in. and under	4 or 5	5 or 6	6 or 7	32 oz. or 35 oz.
1½ in. " " or 1 in. uniform	4 or 5	5 or 6	6 or 7	
2 in. " " " 1½ in. "	.....	6 or 7	7 or 8	
2½ in. " " " 1½ in. "	.....	.....	8 or 9	
Ore, stone, over 100 lb. per cu. ft.:				
1 in. and under	5 or 6	6 or 7	7 or 8	35 oz.
1½ in. " " or 1 in. uniform	5 or 6	.....	.....	
2 in. " " " 1½ in. "	.....	7 or 8	8 or 9	
2½ in. " " " 1½ in. "	.....	8 or 9	9 or 10	
Continuous Buckets	**Projection, inches**			
	Up to 7	Up to 10	Up to 12	
For materials not over 100 lb. per cu. ft.	6 or 7	7 or 8	9 or 10	35 oz.
For material over 100 lb. per cu. ft.	7 or 8	9 or 10	11 or 12	

Fig. 20-1 shows bolts used to fasten rubber lagging to the rim of a grain-elevator head pulley. Some are bent by the pull on the lagging, and the heads are worn off by the slip and creep of the elevator belt. Wearing away the metal causes damage to the pulley side of the belt which adds to that caused by slip and creep.

If the choke is so bad that the belt stops, it is important that the pulley should stop also. In modern elevators with separate electric motor drive an overload release device can be used to throw off current and stop the motor if the load becomes too great; but in old-time elevators, driven from line shafts, it has often happened that a head

pulley continued to turn when the belt was stopped by a choke and the belt was worn in two by the pulley and fell down the legs. Sometimes the damage was more serious; the friction from the revolving pulley set fire to the belt, the flame caused a dust explosion in the elevator casing, and from that followed loss of life and a fire which caused the total destruction of the building and its contents (see page 399).

Belt Creep; Belt Slip. When an elevator belt is so heavily loaded that the head pulley will not drive it the belt may stand still while the pulley turns within the loop. The slip may then be called 100 per cent of belt travel, because the travel of the belt has fallen to zero. If the load is less, the slip is less, but it is always there to some degree, although it may amount to only a fraction of 1 per cent of belt travel when the load is light (see also Chapter 5).

Belt creep is different. In Fig. 22-1, representing the head of an elevator, the up belt under tension T_1 assumes a certain stretch, so that a section of belt normally 12 inches long under no load may be pulled to 12⅛ inches long just as it reaches the pulley. A second or two later that section of belt has reached the other side of the pulley, the tension has fallen to T_2, and the length is only 12 1/16 inches—that is, in passing over the pulley, every foot of elevator belt shortens 1/16 inch, and if the belt speed is 640 feet per minute, the creep or relative movement between the belt and the rim of the pulley is 40 inches per minute or 114 miles per year of average service.

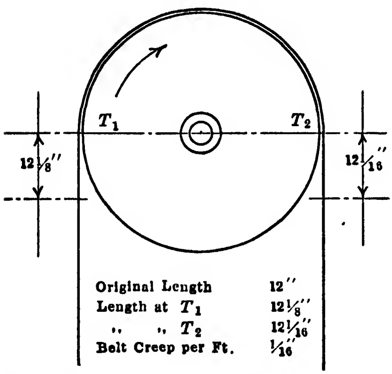


FIG. 22-1. Creep at Head of Belt Elevator.

An illustration of belt creep suggested by W. W. Bird (*Transactions A.S.M.E.*, 1905) is to stretch an ordinary elastic band as a belt over two small pulleys of equal size. If the band has any resistance to overcome, the driven pulley may turn only half as fast as the driver. Then the 50 per cent loss in transmission is not due to slip at all; it is due to the creep of the highly elastic band on the pulleys.

The relations between slip and creep and load, as observed in some of Wilfred Lewis's experiments, are given in Table 22-A, page 361. Column 5 states the combined slip and creep in percentage of belt travel as registered by the apparatus used in the tests. In column 6 Barth has calculated the creep alone from the observed stretch of leather belts

under load. In Lewis's experiments no tests were made of rubber or other fabric belts, but so far as creep and slip are concerned, it is probable that the differences between leather belts and fabric belts would be in degree only, and not in manner. Assuming that the differences are not great, we can take an example from tests 128 and 131, Table 22-A. Here the ratios of $\frac{T_1}{T_2}$ are between 2 and 3, as in general elevator practice, and the tension in the belt, 511 or 557 pounds per square inch, corresponds to between 30 and 35 pounds per inch per ply of fabric, which is not unusual in elevator belts. Here the combined slip and creep (see column 5) amounts to 0.5-1.1 per cent of belt travel, and by calculation practically all of it is creep. It may be said then that in elevator practice a creep of at least 0.5 per cent is unavoidable.

If the load on the elevator belt is increased, T_1 would increase and the belt might slip, unless T_2 were increased by putting more tension on the belt by screwing down or loading the boot shaft. Then with a clean belt, the coefficient of driving contact might rise slightly (see experiment 133, column 7), and the head pulley would drive the belt, but with 0.5 per cent slip plus 1.3 per cent creep. This sum represents a loss of nearly 2 per cent of belt travel; it shows what may happen in an elevator severely overloaded.

Effect of Creep and Slip. It is doubtful whether the coefficient of belt friction in the instance mentioned above would increase in the same proportion in a wet or dusty elevator; but in any event, the slip is objectionable. It means not only a loss of elevator capacity, but, what is more injurious, a movement between the pulley rim and the belt, which, under the pressure due to the load at the head of the elevator, tends to wear away the pulley side of the belt, or the lagging of the pulley, if it is rubber-covered.

Lagging on head pulleys does not last as long as the elevator belt; the two materials are usually of the same class of belting, but the wear on the belt is spread over perhaps 100 to 400 feet of length while the wear on the pulley covering is confined to 10 to 25 feet of lagging. Besides, the lagging is often worn and torn by the heads of the bucket bolts projecting beyond the pulley side of the belt, especially when the bolts are loose or the belt worn. The wear is also greater when the elevator belt is of poor quality or too thin for the work; a belt that stretches greatly under load will creep more on the head pulley than a firm belt, not overloaded, from which the excess elasticity of the fabric has been removed during the process of making the belt.

Pulleys with rough rims, or rims not turned, or rims cast with depressions or slots parallel to the shaft have been used in elevators to increase the grip on the belt. It is doubtful whether the coefficient of belt contact on such a pulley is any greater than on a smooth rim, but there is no doubt that when a belt creeps, as it must, or when it slips, as it may, it is sure to be damaged by the rough rim. If the elevator is overloaded, so that the belt stops while the pulley continues to turn, a rough rim may ruin a belt in a few minutes. The same thing may happen if the elevator is started under load and with a choked boot, as a result of an enforced shutdown caused by failure of electric power, or by some mishap to the elevator machinery or its accessories.

In the western mining country, belt elevators are in general use for handling pulps and wet ores in the process of concentrating. When the mixture is largely water, the spill and splash at the head and the material in the boot keep the belt and pulleys wet, the belt slips on the head pulley and is worn on the pulley side. Experiments have been made with devices to prevent slip, such as slatted-faced pulleys or rough-rim pulleys, but they add to the wear on the belt for the reasons stated above. The real cure for slip is a strong belt, more tension in the belt, and, if possible, the use of an automatic foot take-up to maintain that tension. On this point, see page 401.

Patents have been issued on pulleys with projecting cleats or buttons and on pulley lagging with a roughened face. None of these devices would work at the head of an elevator. The belt must creep and it may slip; any bodily interference with these natural movements will injure the belt.

Elevator head pulleys are sometimes made split, that is, in halves, for convenience in handling or getting on or off the shaft; but when they are made in the ordinary way with the heads and nuts of the clamping bolts bearing against rough unfinished surfaces the bolts tend to work loose under vibration and heavy loads. For important work, split pulleys should have finished bearing pads around the bolt holes at hub and rim. Some engineers will not use split pulleys at all, but specify clamp hub pulleys fitted with not over 0.002-inch clearance to a shaft turned to accurate size. When standard grade pulleys are put on cold-rolled or turned shafting of commercial grade the clearance may be 0.005 inch or more, and very often the pulley works loose and shifts on the shaft in spite of keys and set screws. Some engineers have their head pulleys bored a few thousandths smaller than the accurately turned shaft and then pressed or driven on; others use clamp hub pulleys, heat the hub bolts nearly to redness, put them in place, tighten the nuts, and let the contraction of the bolts clamp the hub

even tighter and prevent the nuts from loosening. In some large ore elevators (Ohio Copper Company, Lark, Utah, *Engineering and Mining Journal*, Vol. 99) the head pulleys are 60 by 38 inches, split, bored $5\frac{7}{16}$ inches with 2 keys at 90° apart, and the outside of the hub at each end is turned to take a $\frac{5}{8}$-by-5-inch welded steel band shrunk on. These pulleys stay tight; to remove them, the bands must be cut apart.

Lagging consists usually of 3- or 4-ply rubber belt fastened on by $\frac{1}{4}$-inch flat-head bolts. It is important that the bolt heads should be below the surface of the lagging; for this reason the pulley rim should be countersunk around the bolt hole to allow the belt to be depressed (see page 152). Pulley faces up to 18 or 24 inches can be covered by a strip of belt of that width; for wider faces and heavy crowns it is easier to use two strips of belt of half the width.

An improved pulley lagging is described on page 152. It is more durable than lagging made of ordinary belt and is far superior in elevator service. It resists abrasion better and is not so likely to be injured by projecting heads of bucket bolts. On the wear of lagging, see page 151.

Face, or width, of head pulleys is usually 1 or 2 inches more than belt width, up to 30 or 36 inches and 3 or 4 inches more for wider belts. The excess of face permits the belt to run out of center for an inch or two, in case the head shaft is not leveled properly or gets out of level by shrinkage or settling of supports.

The crown of standard transmission pulleys is $\frac{1}{8}$ inch on the diameter per foot of face—that is, the face of a pulley for a 12-inch belt is $\frac{1}{16}$ inch higher in the middle than at the edges. Usually elevator head and foot pulleys are made in the same way, but some engineers prefer a crown of $\frac{3}{16}$ inch per foot or even $\frac{1}{4}$ inch per foot for the head pulleys of elevators that are hard to keep in alignment. When the belts are wide and the buckets are in single row the extra high crown may be objectionable (see page 356), but this is not so when the buckets are bolted to the belt in two rows (Fig. 21-4).

Flanged pulleys should never be used on elevators, either at head or foot. Contact between the flanges and the edges of the belt will cut or wear off the edges.

CHAPTER 23

ELEVATOR BOOTS

Purposes of Elevator Boots. An elevator boot serves several purposes:

1. To confine the material to the path of the buckets.
2. To support the foot shaft.
3. Generally, but not always, to support the elevator casing.

There are two general types of elevator boots—boots with fixed shaft bearings and boots with take-up bearings.

Fixed Bearing Boots. Where elevator pits are small, or hard to get at, or blocked at times by spill from feeding conveyors, or by chokes in the elevator itself, it may be an advantage to use boots with fixed bearings and put the take-up bearings at the head of the elevator. This is standard practice in some cement mills; the elevators are provided with stairs and platforms which make it convenient to reach the head to adjust the take-ups. The take-up screws in this position are comparatively clean, but in a deep pit they become dirty and hard to turn in spite of dust guards and ordinary efforts to keep the pit clean. In some mills, pits are never permitted to get dirty with spilled material; but in others, with the class of labor employed, the great volume of material handled, and the 24-hour continuous operation, it is practically impossible to keep them clean at all times.

Fig. 23-1 shows a pit for the foot of an elevator for crushed cement rock ¼ inch and under. The entire structure below the floor line is concrete; the foot bearings rest in openings in the 8-inch walls forming the sides of the boot. Sheet-steel doors on each side give access to the interior of the boot, and a chute with a slide door permits spilled material or material which has been cleaned out of the boot to be shoveled from the pit into the path of the buckets.

There are other reasons for using boots with fixed bearings. In a take-up boot, with the wheel in its upper position, there is a mass of material lying beyond the sweep of the buckets which may pack so hard that unless it is dug loose by hand before the wheel is set down to a new position the buckets will not pick it up, but will be torn loose

or pulled out of shape. This is true of some chemicals and fertilizers, and of cement; for these, a boot with a definite sweep of the buckets is best. Fixed bearing boots are also used for food products which spoil if allowed to accumulate beyond the sweep of the buckets, also in elevators which at different times handle different materials which should not be mixed.

If the material handled is coarse, hard and lumpy there is some advantage in using a fixed-bearing boot, so that the pieces pushed by the buckets slide over a curved steel bottom plate at a fixed distance from the foot shaft and do not drag over a bed of lumps lying beyond the sweep of the buckets. There is, however, a limitation to this; if the material is approximately round or cubical in shape, buckets can be run close to a bottom sheet, but if the pieces are sharp-cornered, or

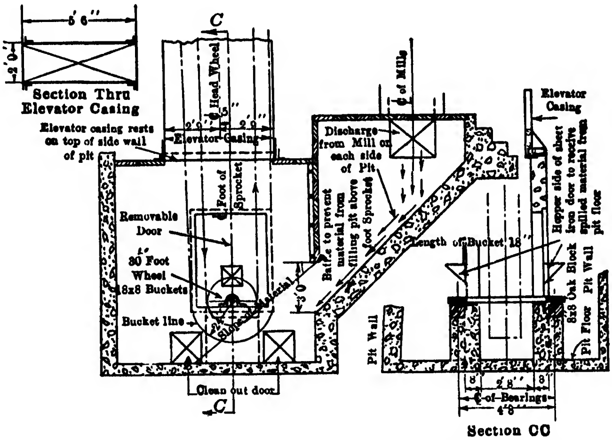


FIG. 23–1. Concrete Pit and Boot for Pulverized Stone Elevator.

angular or long, like slivers, there is danger that they may wedge between the buckets and the bottom sheet and either damage the belt or chain or tear the bucket loose or punch a hole in the sheet. If these materials are handled in a centrifugal discharge elevator, which is not always the best way, then the clearance should be larger than the dimension of the lump and the bucket should be large enough and strong enough and the belt or chain that carries it sufficiently powerful to dig the material as if from its own bed.

In many elevators for sand, ashes, coke, and similar abrasive substances the boot is made as the lower part of the elevator casing and not as a separate piece; in these elevators it is not customary to use a curved-bottom sheet, or even a flat-bottom sheet; the floor on which the casing rests forms the bottom. This construction saves something

in first cost and also in renewals, for a curved boot bottom does not last long when the elevator handles lumpy and abrasive materials.

In some elevators even less attempt is made to confine the material to the path of the buckets; in hot-clinker elevators, for instance, as used in cement mills, the boot consists of a trench or pit with the shaft bearings mounted on the side walls or on cast-iron pedestals between the walls. The clinker then forms its own boot under and alongside the foot wheel. Most of it is like sand or gravel in size, and even if a large fused mass of clinker or a portion of dislodged kiln-lining too large for the buckets should slide down into the pit, it will not jam there, as the buckets can tumble it out of the way. These clinker elevators are chain machines, but the arrangement described is applicable to some belt elevators as well.

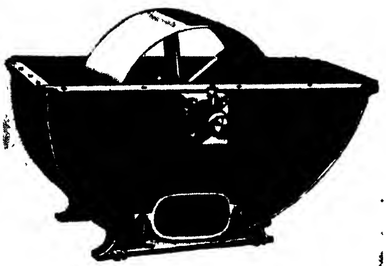


Fig. 23-2. Cast-iron Fixed-bearing Boot. (Jeffrey Manufacturing Company.)

Manufacturers' designs of fixed bearing boots are of steel plate throughout or with cast-iron sides and a steel-plate bottom. Fig. 23-2 shows one which can also be furnished with a sectional cast-iron bottom to take the wear from abrasive materials. Fig. 23-3 shows a boot for fine crushed rock, designed by a cement company for its own use; it is fed at front and back. The ¼-inch bottom sheet is not fastened in place, but is held in a $\frac{5}{16}$-inch slot between a bent angle and a bent flat fastened to each side of the boot. The shaft bearings can be easily removed; a retainer ring on the inside of the boot holds the heads of the bolts from turning and keeps them from falling out of place.

Designers of elevators sometimes choose fixed bearing boots so that the elevators can be driven at the foot without interfering with the take-up. This is perhaps the poorest reason for using such boots. Belt elevators carrying light stuff, like wood chips, have been driven at the foot, but the head take-ups need constant attention to keep the belt in driving contact with the foot pulley. In chain elevators driven at the foot the chains slip and break

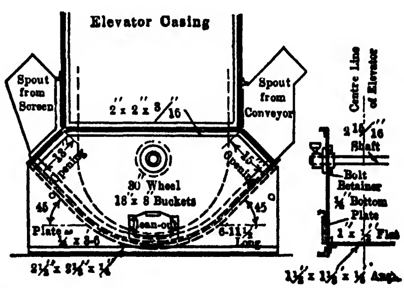


Fig. 23-3. Double-sided Boot for Pulverized Cement Rock.

unless they are kept at a steady tension and free from slack. Experienced engineers do not drive elevators at the foot.

Take-up Boots. Over thirty styles of take-up boots are sold by American manufacturers, and most of them are made in a number of sizes. There are two principal kinds—boots for coal and similar coarse, heavy substances, and boots for grain. The distinguishing features of these boots are determined by the pick-up of material; for a discussion of this, see pages 303 and 305.

Boots for coal and similar coarse, heavy materials consist of a pair of sides, usually cast iron, joined by a curved bottom plate, usually of steel plate. The front is sloped to direct material into the path of the buckets and at the same time to form a clearance space into which the pieces may be pushed instead of jamming against the bottom sheet, as might happen if the bottom were fully rounded with the lower position of the shaft as a center.

Fig. 23-4 shows a section through one side of a standard boot. The bearings, closed at the outer end to prevent loss of lubricant, are made with a spherical center so that the shaft will not be cramped even if the two bearings are not adjusted alike, or if the shaft is not square with the sides of the boot. A rectangular slide frame holds the bearing and moves up and down in a slot in the side of the boot; a cap for the slide frame allows the bearing to be removed easily. There is a steel plate on the inside of the boot which travels with the shaft and keeps the slot closed in any position of the shaft. Springs acting on clamp bolts keep the plate tight against the boot side, prevent dust from coming out, and keep material away from the bearings and the slides. The screw works in a nut held in a recess in the boot side by a separate yoke piece, a small casting which will break and save the boot side in case of a choke or an accident in the boot. An opening, or clean-out, in each side of the boot, is closed by a cover, recessed to form a dust seal, and paneled inwardly so that it comes flush on the inside of the boot and does not leave a cavity or depression in which pieces of material might be jammed and then tear buckets off or damage the belt. This feature is not important in a grain boot, but it is of value in a boot that handles hard, coarse material.

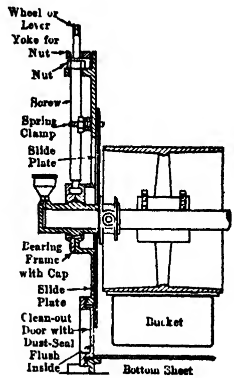


FIG. 23-4 Cross-section through One Side of a Boot for Coal and Minerals. (Link-Belt Company.)

Dust-tight Boots. Other boots, in styles similar to the above, have removable cast-iron bottom plates for greater durability; others, made to be dust-tight, have covers outside the bearings in addition to the inside slide plates (Fig. 23-5). The jaw levers pivoted to the top of the take-up screws give a large leverage for adjusting the bearings and do not interfere with the elevator casing as a hand wheel would.

Boot Take-ups. When boots are made as the base sections of steel or wood elevator casings the take-ups are necessarily separate parts bolted on. Fig. 23-6 shows one style made with a slide plate to keep the slot in the side of the boot closed at all times.

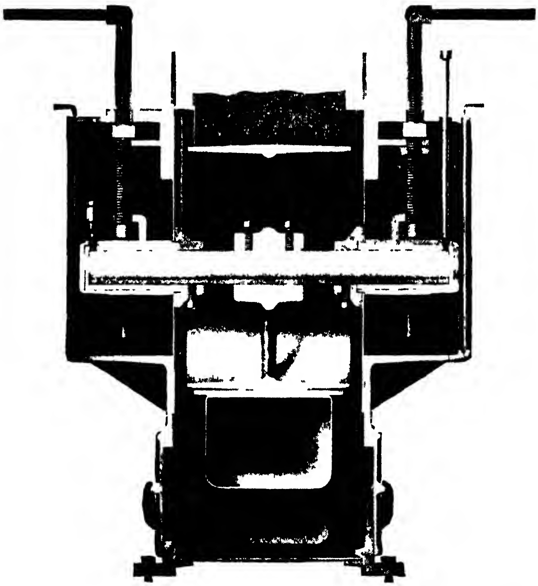


FIG. 23-5. Take-up Boot with Outside Dust Seals.

When a boot works in a damp or wet place it is advisable to make the take-up nuts of brass, so that the take-up screws will not rust tight in them.

Boots in Missouri Lead Mines. Standard boots listed by manufacturers are well suited to handle coal and other substances which are not too hard or too abrasive. With such materials, the wear on the boot sides and bottoms is not too rapid, but for elevators that carry ores and minerals in ore-reduction plants, concentrator plants, and cement mills, the boots are often of special design, with the idea of resisting abrasion by making the sides and bottoms very heavy, or else dispensing with iron and steel construction entirely. In the lead-mining district of

Missouri, crushed ore, in size from sand up to 3/8 or 1/2 inch, is handled wet or dry in belt elevators with boots that consist of a rectangular box of planks or a concrete pit which in width has a foot of clearance on each side of the pulley, and in length has room enough for a man to get in and use a shovel behind the pulley. These boots may be 6 or 8 feet below the floor of the mill. The tension is put on the belt and slack taken up, not by screws, but by the arrangement shown in Fig. 23-7. A 4-by-6-inch vertical post is notched at the bottom end to take the shaft, and the shaft turns in the notch. A pivoted lever, held in posi-

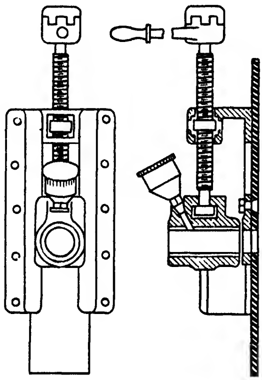


FIG. 23-6 Take-up with Dust Seal for Steel or Wood Boot.

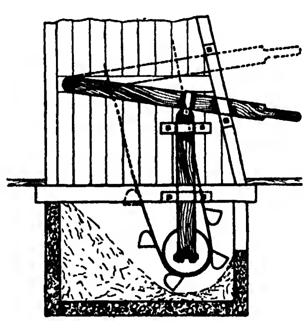


FIG 23-7. Boot for Crushed Ore, Joplin, Missouri.

tion on each side of the wooden casing by wedges or clamp bolts, acts down on the 4-by-6-inch post and applies pressure to the ends of the foot shaft. In some of these elevators the shaft has a flanged sleeve set-screwed on it at each end to take the wear from the pressure post; when the sleeve wears out it is thrown away and a new one is slipped on. No attempt is made to lubricate it.

Take-up by Moving the Whole Boot. Fig. 23-8 shows a boot that has a take-up arrangement and yet maintains a fixed clearance between the sweep of the buckets and the bottom plate. The elevator handled crushed dolomite; it had to be inclined on account of the position of the bin which it fed; and because of the small space available, the casing had to be very narrow with the buckets running on guide angles. It was necessary to prevent an accumulation of material in the boot, and since take-ups could not be placed at the head, the boot was therefore mounted on a frame pivoted at one end and adjusted by two bolts to give the necessary take-up travel.

Position of Feed. From what has been said in Chapter 17 it is clear that in a centrifugal discharge elevator the buckets take their load on

the rising side of the foot wheel, and at the ordinary speeds of Table 17-A and 17-B, the buckets cannot take a full load unless they receive some material at or above the level of the foot shaft. This is especially true if the foot wheel is smaller than the head wheel. How this affects the delivery of grain to buckets in a high-speed grain elevator has been shown in Figs. 17-6 and 17-7.

When a centrifugal discharge elevator takes coal or some similar material from a sloped-front boot the action is very much the same. Fig. 23-9 shows a standard form of boot with a front sloped at 45° and with the lower edge of the feed chute about level with the center of the foot wheel in its upper position of take-up travel. When the elevator starts, and material is fed into the boot, little or none of it is caught by the buckets until a bed of dead material forms beyond the sweep of the buckets and piles up in the front of the boot. If the material is fine and dry and stands at a low angle of repose, the bed of dead material will not reach up into the chute; if it stands at 45°, as shown in the figure, it will partly block the chute; if it is damp and sluggish like foundry sand or boiler-house ashes, or bituminous coal or other material received in open cars and exposed to rain or snow, then the angle of repose may be so steep that the chute will be choked and the flow of material stopped.

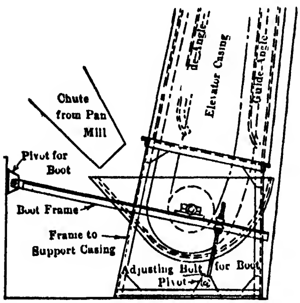


FIG. 23-8. Take-up Adjustment by Moving the Whole Boot.

When the wheel is in the lowest position of take-up travel (Fig. 23-10) the boot is kept clear of accumulated material, and the disturbance of the material beyond the reach of the buckets on the rising side of the wheel prevents it from blocking the chute.

There are standard forms of take-up boot with steep fronts, the angle being more than 45°. Fig. 23-11 shows one with a 55° front and with the lower edge of the feed chute *above* the center of the foot wheel in its upper position of take-up travel. When the wheel in such a boot is all the way up, the bed of dead material beyond the sweep of the buckets will not back up in the chute unless it stands at a steep angle, and hence this boot is likely to work better than a low-front boot if the material is damp, sluggish, or apt to pack hard and stand on a steep angle of repose. When the wheel is in its lowest position (Fig. 23-12) there is even less chance of a choked chute, and the buckets fill well.

One drawback of a steep-front boot is that, when the take-up is all the way down, an excess of feed over elevator capacity may cause the

foot wheel to be buried so deep that the buckets cannot dig themselves free. This may happen if a heavy load is dumped into the boot or if the elevator should slow down or stop for any reason while the feed continues. The line *A—B* in Figs. 23-10 and 23-12 shows how the material may pile in the two styles of boot under such circumstances.

Summary. Boots with fronts sloped at 45° (Figs. 23-9 and 23-10) have a low chute, save some depth of elevator pit, work well with dry,

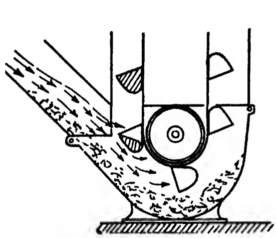


FIG. 23-9.
45° Slope Boot, Wheel in High Position.

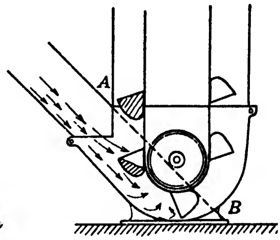


FIG. 23-10.
45° Slope Boot, Wheel in Low Position.

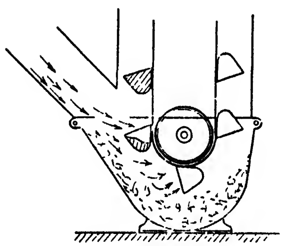


FIG. 23-11.
55° Slope Boot, Wheel in High Position.

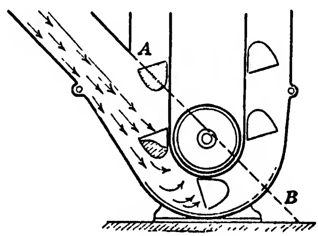


FIG. 23-12.
55° Slope Boot, Wheel in Low Position.

free-flowing materials, but not so well with sluggish materials unless the foot wheel is set low and the wheel and buckets are of a size that makes the sweep of the buckets come close to the bottom sheet. Boots with fronts sloped at 55°, 57½° or more are higher in front, require some inches more of depth in the boot pit, work well with all materials fit to handle in centrifugal discharge elevators, but require free-flowing substances to be fed with some care, so as not to swamp the foot wheel when it is in a low position of take-up travel (see Figs. 23-11 and 23-12).

Fly-feed or Scoop-feed. In an elevator with continuous buckets run at comparatively slow speeds it is possible to chute the material directly into the buckets, but in centrifugal discharge elevators, where

the buckets are spaced apart on a belt or chain and travel at the speeds given in Table 17-A or 17-B, there is little or no gain in an attempt to deliver the material directly to the buckets, the so-called "fly-feed." If the lip of a chute set for "fly-feed" is at or below the level of the foot shaft it is inoperative, because the material will be thrown out of the buckets by centrifugal force (see page 302) and will drop down and be scooped up when enough has accumulated under and in front of the foot wheel. If the chute is higher and directs the stream of material above the center of the wheel into buckets moving at a speed of 5 to 10 feet per second, only a part of it is caught, most of it falls down into the boot, and some may be splashed and scattered so as to fall on the top of the foot pulley. Material caught between the pulley and the down belt wears or cuts the belt, and with the "fly-feed" the other side of the belt suffers some injury from the direct impact of material striking it between the buckets.

The best feed for hard, gritty, lumpy materials handled in centrifugal discharge elevators is the "scoop-feed" where the buckets act on a yielding mass fed in on the sloping front of a boot, or where the delivery is such that the material can pile to form its own boot. The slope of the front of the boot and the height of the chute with reference to the foot wheel depend on the nature of the material, whether it is free-flowing or sluggish (see page 305).

Feeding into Side or Back of Boot. Since buckets in centrifugal discharge elevators never pick up material below or behind the foot wheel, feeding into the boot at those places merely adds to the work of the elevator belt and buckets by the power required to drag the material over the inside surfaces of the boot or over a bed of dead material from the place of feeding to the front of the wheel where the buckets pick up their load. In grain elevators this does no particular harm to the belt or buckets because the grain moves freely and causes very little wear, but in elevators handling lumpy or abrasive material the bottom sheet and side plates of the boot, and also the lips of the buckets, are likely to be worn by the friction of the material dragged around beneath the wheel. A more serious matter is the chance that pieces may wedge fast between the buckets and the bottom sheet, or between the buckets and the bed of dead material lying beyond the sweep of the buckets, with the result that buckets are torn apart or pulled off the belt, and with consequent injury to the belt itself.

It is often convenient to feed an elevator from the back or side to save depth of pit and height and length of feed chutes. When the material is in small pieces this is permissible, but it is dangerous to handle hard and lumpy material in this way.

Where the foot of a grain elevator is lowered into a cargo of grain the foot frame of the "marine leg" must be open at sides and back to feed to the buckets. Fig. 23-13 shows such an open frame; when it is in operation the most active flow of material is at the back where the buckets go down.

Size of Foot Pulleys. It is a common mistake to make these pulleys too small. The object may be to save space, or to avoid a few inches greater depth of pit, or to save something in the price of a boot.

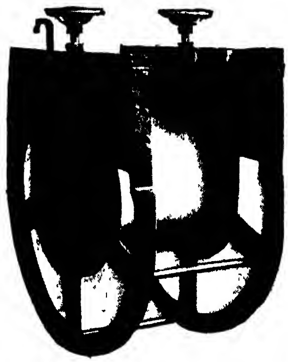


FIG. 23-13. Open Boot or Foot Frame of Marine Leg.

So far as the belt is concerned, the general objection to a small pulley is that it tends to separate the plies of fabric in a built-up belt by stretching or breaking the bond which holds the plies of fabric together, whether that bond is stitching, as in canvas belts, or some cementing compound, as in rubber or balata belts, or the binder threads in a solid-woven belt. In conveyor practice, foot pulleys have seldom less than 3 or 4 inches of diameter per ply of belt. Belt elevators are generally shorter between centers than belt conveyors, and hence the belt bends over the pulleys oftener for the same belt speed. For that reason, foot pulleys of belt elevators should be at least 4 inches in diameter for each ply of belt—that is, a 24-inch pulley for a 6-ply belt. If the elevator is short, the belt bends oftener, and a ratio of 5 to 1 is better still, to prolong the life of the belt and postpone the day when the belt will fail by separation of the plies. It must be said, however, that most elevator belts are not discarded on account of the internal wear which causes the plies to separate; some belts handling clean, dry substances like grain may die of old age, but most elevator belts succumb to external injuries which have no connection with the diameter of the pulleys they run on.

The great objection to a small foot pulley is the bad pick-up. For reasons stated in Chapter 17 and shown in Figs. 17-3 and 17-5 a small foot pulley may not permit the buckets to pick up any material until they are on the straight run above the pulley and away from the influence of centrifugal force. When that happens, the buckets, while they are in contact with the foot wheel, do nothing but stir up material uselessly, wear themselves out, and perhaps injure the belt.

The larger the foot wheel in comparison with the head wheel of a centrifugal discharge elevator, the less the influence of centrifugal

force in the boot, the lower the buckets pick up their load, and the less the energy wasted in the boot.

Large foot pulleys are better than small ones in another respect; they support the buckets better when the buckets pick up their load. Fig. 23-14 shows, to scale, a bucket with 6-inch projection on a 6-ply belt running on a 15- or 27-inch pulley. Since the bolt fastening is not rigid, the bucket moves in the direction of the arrow when it meets some resistance from the material, thus exerting a pull on the bolts until the bucket finds a backing against the belt at some point *A*. The larger the pulley, the less the bucket will move and the smaller will be the

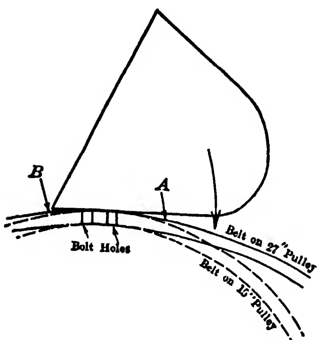


Fig. 23–14. Better Support of Bucket on Foot Wheel of Large Diameter.

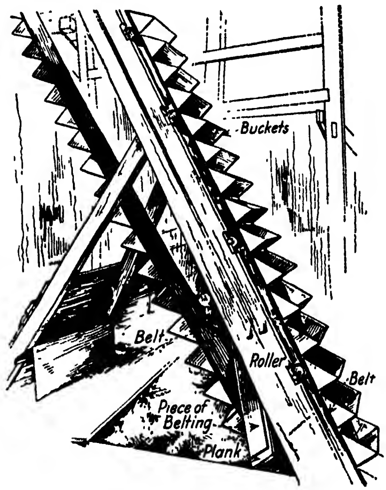


Fig. 23–15. Deflector for Spill on Return Run of Elevator.

pull on the bucket bolts. It will also gap away from the belt less at *B* and bits of material will not be so likely to catch and stick there and injure the belt.

Material Catching between Belt and Foot Pulley. More elevator belts are injured from this cause than from any other. The pieces of material which cause the trouble may have been spilled from the buckets on the vertical run or on the head wheel; or lumps may fall from the head chute on a rebound, or when the material backs up in the chute from a choke or when the bin it supplies is full. When the feed chute is too high with reference to the foot wheel, or when it is set for "fly-feed," this trouble is more likely to occur than when the chute delivers to a sloped-front boot of proper design.

Guards placed close over the pulley do not cure the trouble altogether. If placed so close to the pulley as to shed falling pieces, they

may serve to confine material which has gotten through the clearance space which must be left between the guard and the belt, and thus do more harm than good. If the elevator is inclined instead of vertical, spill from the head is less likely to fall on the foot pulley, especially if the casing is made, as in Fig. 23-7, with a vertical face on the descending side.

In some inclined elevators handling large pieces of stone, guards or shields have been used with success to deflect from the foot pulley any pieces of stone which may fall or roll down along the inside face of the descending belt. Fig. 23-15 shows such a device; it consists of a heavy plank set diagonally across the elevator above the foot pulley and provided on its lower edge with several thicknesses of old belting which project far enough to keep close to the elevator belt and yet yield when the belt vibrates or sways, as it always does.

Devices like this need the intelligent interest of the men in active charge to keep them in working order. The plank may need adjusting from time to time, or the strips of belting may have to be replaced when worn. With such attention, a guard like the one shown may prevent serious injury to the belt and may pay for itself many times over in the longer life of the belt; without such attention, it may be discarded as a failure.

Elevators with No Foot Pulley. The chances of injuring the belt would be fewer if belt elevators could be run without foot pulleys. This has been done in the works of a copper company in Arizona, under the Cole patent of 1920 (Fig. 23-16), which covers the use of a belt elevator without a foot pulley but with a normally slack belt guided by a pulley on the rising side above the feed point. Some of these elevators had 24-inch belts about 10-ply with 16-inch buckets set staggered. "When running at the speed for a proper dumping effect for the buckets the lower end of the belt loop would take a form almost the same as that of the boot pulley and would take up the feed as well as if the usual boot pulley were used. The snub pulley would only come into play when an excess of feed was fed to the machine. There was no strain on the belt other than that of carrying the load; no sand or quartz particles were forced into the cover of the belt, the latter being clean all the time." *

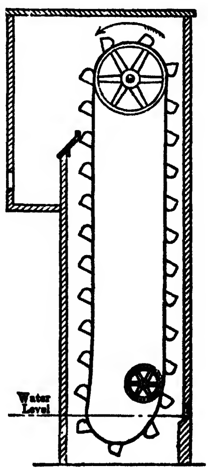


FIG. 23-16. Crushed Ore Elevator with No Foot Pulley.

* Communicated to F. V. Hetzel by Mr. David Cole.

In many elevators the load is so heavy that take-up tension must be added in order to get sufficient driving effect (see page 364). The foot pulley is then a necessity.

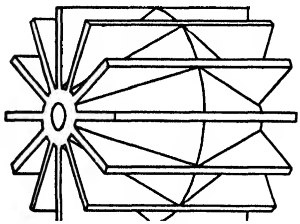


FIG. 23-17. Foot Pulley with No Rim.

Special Foot Pulleys. Several forms of pulley have been used to lessen the risk of injuring the belt. The Boss pulley shown in Fig. 23-17 presents less surface to the belt, and if the material is fine and dry it will be shed to each side of the belt by the conical surfaces in the pulley. Flanged pulleys in two sections have been used (Fig. 23-18); they offer less surface to the belt, but, since the pulley has no crown, flanges are necessary to guide the belt. The flanges are bad because the edges of the belt tend to ride up on them and are worn, and the wheels are likely to act as a trap for material rather than a relief for the belt.

Neither of these devices is of use when the material handled is in pieces that will wedge in the pulley. A piece of sharp stone wedged between the halves of the flanged pulley or between the arms or vanes of the Boss pulley may go around many times and hurt the belt in many places, while the same piece might go once around an ordinary pulley and then fall off.

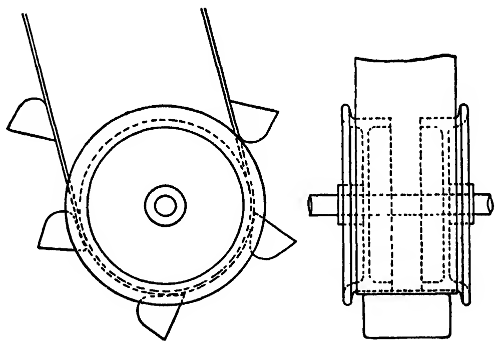


FIG. 23-18. Double Flanged Foot Pulley.

It is possible that foot pulleys for thick and heavy belts on elevators handling ores and stone might be made with pneumatic or cushion rims that would be more yielding than the belt and still be rigid enough to guide the belt and apply tension to it.

The problem is a serious one, and one that is hard to solve. Men in the business have seen 10-ply belts cut clear through by stone jammed against the foot pulley; the force that will do this is hard to control by mechanical means. At present it is not attempted; the burden is put on the belt manufacturers, and they are making belts thicker and heavier, with outer covers and internal cushions of rubber to resist the blows and the pressure.

Ordinary foot pulleys for elevators handling coal and other materials which are not too abrasive are generally made in "double-belt" weight which, for the usual range of diameters, from 15 to 30 inches and faces 6 to 18 inches, means that the edge of the rim is $\frac{1}{4}$ to $\frac{5}{16}$ inch thick at the edge and $\frac{3}{8}$ to $\frac{1}{2}$ inch thick in the center. The crown is usually standard—that is, $\frac{1}{8}$ inch on the diameter per foot of face. The face measures $\frac{3}{4}$ to 2 inches wider than the belt for the sizes used in elevator boots.

In handling sand and ores, the stir of material in the boot and the fine stuff adhering to the belt combine to wear out the rims of ordinary foot pulleys. For this service it is economical to order pulleys with rims thicker than standard, and with heavy arms, or with plate centers instead of arms.

Fastening Foot Pulleys to Shafts. On this subject, see page 394.

Amount of Take-up Travel. Take-ups for belt conveyors are often made with 3 feet or more of travel, so that 6 feet or more of belt stretch can be removed without cutting and resplicing the belt. Elevator belts are generally shorter than conveyor belts and the stretch is less, but aside from that the range of travel of the foot shaft in a take-up boot must be limited between positions at which the buckets will pick up material properly and at which the feed chute will neither be choked nor yet swamp the foot wheel (see page 383).

This circumstance limits the take-up travel in grain boots to 12 or 15 inches and in other belt elevators to 6 or 8 inches. In chain elevators the take-up travel should be at least enough to permit the removal of one pitch of chain, if the links are all alike; of two pitches if the links are alternately inside and outside, as in some steel chains; and in continuous bucket elevators, at least enough to remove one bucket.

Lubrication of Boot Bearings. Although oil is generally used for the bearings of grain boots (see page 393), grease is preferable for most elevators handling coal, minerals, ores, and rough materials. It is better suited to the cheap and simple bearings of such machines, and often it keeps dirt out of the bearings by forming a collar or ring where it squeezes out of the end. The best way to apply grease is to

put the grease cup directly on the bearing (see Fig. 23-4 or Fig. 23-6); if the pit is deep or if one side of the boot is inaccessible, a pipe may be used to lead the grease to the bearing. There is always some uncertainty about lubricating in this way; if the pipe is long, or is bent, or has elbows in it, the grease may clog in it in spite of screwing down the cap of the grease cup. Turning the cap may merely compress air in the pipe, or if the cap is small, it may turn so hard that with some classes of labor it will be neglected.

To make sure of lubrication in spite of occasional neglect, boots are made (Fig. 23-19) with large grease pockets cast on the bearings. The

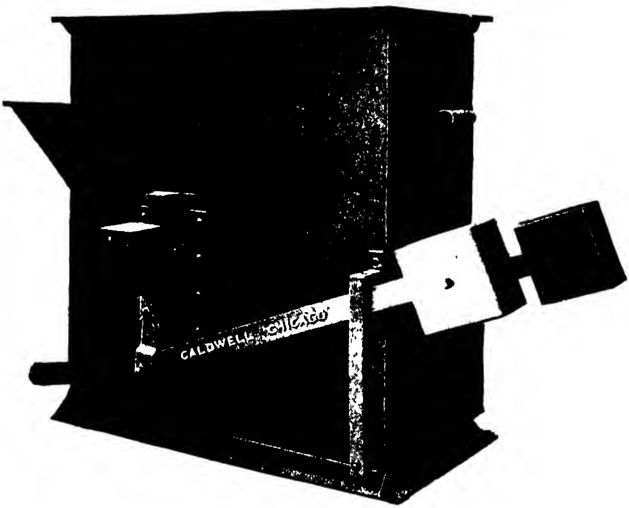


FIG. 23-19. Automatic Take-up Boot with Grease Pockets on the Bearings.

mass of grease rests directly on the shaft and feeds to it by gravity. Other boots have bearings made of hard wood impregnated with oil; these bearings require no lubricant, they are comparatively cheap, and, when worn, they are thrown away. Fig. 23-20 shows an arrangement which has been used in the western mining country. The boot shaft is a hollow brass shaft which contains a supply of oil; wicks feed oil to the bore of the foot pulley which runs loose on the shaft; the shaft itself does not revolve. In elevators handling gritty ores, the pulley side of the belt may be worn off if the foot pulley does not revolve freely. The pulley in an ordinary boot may stick if the two take-up screws are not adjusted alike or if the two bearings bind the shaft because of dirt or lack of lubricant. When the pulley is loose on the shaft, as in Fig. 23-20, it is more certain to turn; there is less

weight to be revolved, and the turning of the pulley is not affected by the way the take-ups are adjusted.

In some elevators no attempt is made to lubricate the foot bearings. In hot-clinker elevators in cement mills the foot bearings are hard to get at, being in a deep pit where the heat is intense. It is good practice here to use chilled iron bearings and run them until they are worn out. This is more economical than to use better bearings and then pay for oil, attendance, and maintenance. A bearing which is not lubricated may screech and get hot, but these faults are not serious at the foot of a clinker elevator. These elevators are chain machines, but the same

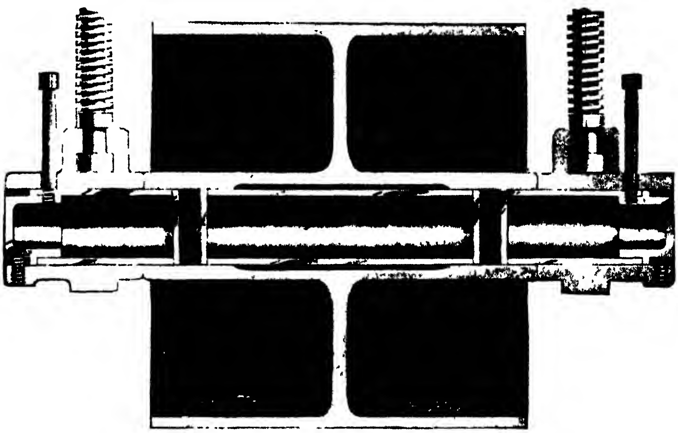


FIG. 23–20. Loose Foot Pulley on Hollow Shaft with Internal Lubrication.

principle also applies to some belt elevators. On this point, see page 381.

Grain Elevator Boots. Old-time boots were of wood, made either as a prolongation of a wooden casing or pair of legs, or as a separate box on which the casing was mounted. The latter construction was more expensive, but the bearings were often mounted in a better way, the interior was more accessible, and the boot was more easily cleaned. Wood boots are still used in some places; but they are subject to decay in damp pits, they get out of shape under the weight of high casings, and there is always the risk of a fire starting in a wood boot from a combination of oil-soaked wood and the heat from a bearing which has not been lubricated properly.

Modern grain boots consist either of the base portion of a steel casing or of a structure of cast-iron and steel plate on which a steel casing is mounted. All-steel construction is always used in modern elevators where the legs are very high, and where, on account of a large

head wheel, the lower sections of the down leg approach the foot at a decided slant. Fig. 23-21 shows such a steel-plate boot with a 30-inch-diameter foot pulley for a 40-inch belt. It is built as part of the casing with $\frac{3}{16}$-inch plate, 2-inch angles, and $\frac{3}{8}$-inch bolts and rivets 4-inch pitch. Two $\frac{3}{16}$-inch removable slide plates give access to the inside of the boot, front and back. The shaft is $2\frac{7}{16}$ inches with 15-inch travel. The elevator is 196 feet center to center. The boot bearings slide up and down in a cast-iron frame bolted to the boot sides, and carry a slide plate which closes the slot on each side of the boot. A weighted frame mounted above the shaft provides an automatic take-up for slack and keeps a tension on the belt.

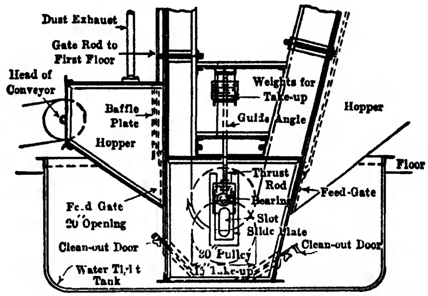


FIG. 23-21. Foot of 40-inch Receiving Leg. Public Grain Elevator, New Orleans. (Ford, Bacon & Davis, Engineers.)

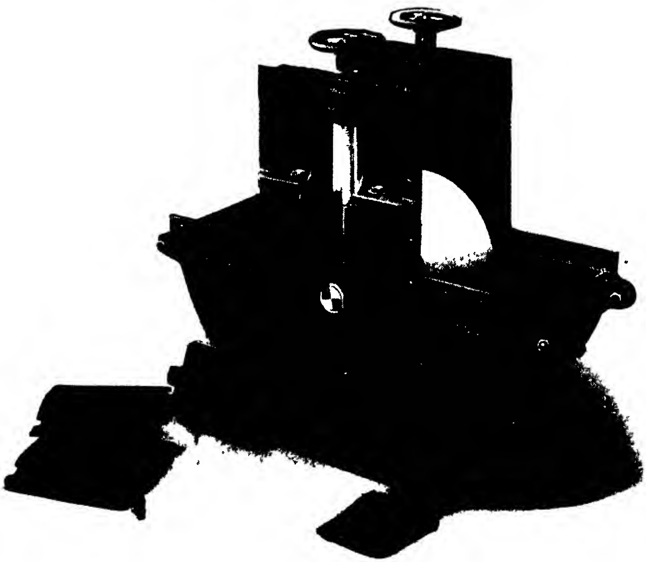


FIG. 23-22. Cast-iron Grain Boot with Removable Bottom Plates. (Used with Wooden Legs.)

Cast-iron Boots. Various styles of grain boots are listed by manufacturers. Generally they have cast-iron sides joined by cast-iron or steel plates curved to form the bottom of the boot. In the best designs,

portions of the bottom plates are removable (Fig. 23-22) to give access to the inside for cleaning or removing obstructions; sometimes only small clean-out doors in the side plates are provided, but these are too small to do much good when the boot chokes. At such a time, the grain is piled higher than the shaft and it is better to get at the trouble quickly by removing a section of the bottom so that a shovel can be used and, if necessary, damaged buckets can be taken off the belt.

Fig. 23-23 shows a heavy cast-iron boot for a large grain elevator with a 24-inch pulley and 11 inches of adjustment. The bearings are closed at one end, and the opening at the other for the $2\frac{7}{16}$-inch shaft is sealed against the entrance of dust and dirt and against the leakage of oil by a felt washer and a packing ring. It carries a supply of oil

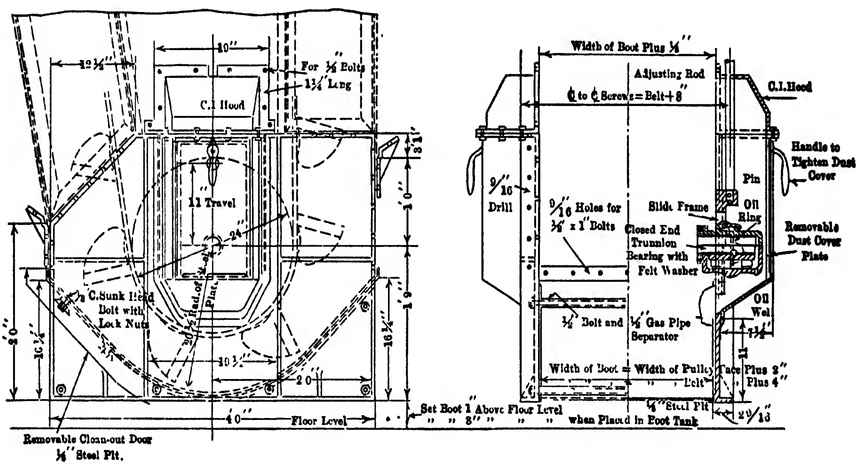


FIG. 23-23. Heavy Cast-iron Grain Boot Used with Steel Casing.

in the base of the casting, enough for several weeks' run. A loose brass oil-ring delivers oil to the shaft, and a hinged cover permits the attendant to see whether the ring turns and the oil reaches the shaft. The trunnions or pivots of the bearing are carried in a slide casting, so that the shaft can be leveled and the belt made to run straight on the pulley independent of the setting of the boot. The slide casting is pinned to the adjusting screw and travels with it, the screw being moved by a handwheel which acts as a nut. The screw does not turn.

Lubrication of Grain Boots. In cheaper boots the adjusting screw turns, is necked at the lower end, and has a loose connection with the bearing itself, and in some of them the oil is fed to the bearing through the screw itself, which is then made of heavy pipe or hydraulic tubing. Experience with this method of oiling is not altogether satisfactory;

dirt gets into the loose connection at the lower end of the screw and plugs the oil hole shut. Boots are often placed in deep and narrow pits, and it may be desirable to oil the bearings from the floor level, but in such cases it is better to use separate oil pipes screwed tight into the bearings and not oil through the take-up screws.

Oil is generally used for the bearings of grain boots, on account of the high speed of the shaft, often 100 r.p.m., and for cleanliness. The objection to grease lubrication is that the dirty collar of grease which accumulates at the ends of the bearings may fall off into the grain and spoil it for flour milling.

In some ways it is better that the attendant should get down into the pit to oil the bearings; he can then see and feel the bearings and make an inspection of the boot. It is a common experience that boot bearings are apt to be neglected; then if the shaft should seize tight in a hot bearing, there is danger that the pulley may stand still and rub off the inner plies of the belt, or if the pulley is fastened only by set screws, it may turn while the set screws gouge grooves in the stationary shaft, or it may shift endways on the shaft until its rim cuts through the side of the boot. These mishaps sometimes cause expensive shut-downs and then costly repairs. In the design of important grain elevators it is well to provide for good lubrication by using well-made self-oiling bearings or anti-friction bearings, to make the pit large enough for access to both sides and both ends of the boot, and to keep the bearings outside the boot where they can be seen and felt. In the large elevator shown in Fig. 23-21 the foot shaft with its pulley, slides, and automatic tension device weighed over 2200 pounds; a load of about 1100 pounds on each shaft bearing at about 100 r.p.m. represents a duty important enough to deserve good bearings, thorough lubrication, and regular inspection. As a precaution against heating, the foot bearings of elevators, as well as other bearings, are sometimes fitted with fusible metal plugs which make electrical contact and sound an alarm when the temperature of the bearing goes above 165° F. or some other established temperature.

Ball bearings or roller bearings on elevator foot shafts avoid many of the uncertainties of old-time babbitted bearings. They reduce the fire risk and save power.

Pulleys in grain boots are always made with a crown face and of the weight known in the trade as "double-belt." In small elevators it is sufficient to hold the pulley in place on the shaft by two set screws, but it is always an advantage, and in important work it is very desirable, to hold the pulley more securely. It is a common complaint that, in conveying and elevating machinery, set-screwed wheels will not stay

in place, but shift on the shaft. The reason is that commercial shafting varies in diameter from its nominal size; it is seldom oversize but often 0.003 or 0.004 inch undersize. Pulleys in the trade are bored and reamed a few thousandths oversize to make sure of their going on commercial shafting; the result is that, when a set-screwed pulley with an oversize bore goes on an undersize shaft with perhaps 0.005 to 0.01 inch clearance, it is very likely to shift, especially if the face is wide and the hub relatively short. Some men in charge of grain elevators (and other elevators) always "spot" the shaft, that is, drill shallow recesses in it to receive the points of the set screws; some use long set screws with lock nuts to keep them tight in the pulley hub; some keep the pulley in place by fastening collars with set screws to the shaft on each side of the pulley hub. A better way for pulleys and shafts of commercial grade is to key the pulley to the shaft with a fitted key and put two set screws over the key. A still better way, although it is seldom used for foot shafts, is to turn the shaft to an exact diameter, bore the wheel a few thousandths small, and then press or drive it on.

Pulleys wider than 20-inch face should have double arms.

Pulleys for large elevators are generally made with closed ends, usually by fastening on steel-plate discs, so that the grain is kept out of the pulley and the arms do not add to the work of the elevator belt by stirring up the grain.

Automatic boot take-ups may be used: (1) as a convenience to avoid attention to take-up screws; (2) to maintain driving contact on the foot pulley or sprocket wheel in those exceptional cases where an elevator is driven at the foot or where power is taken from the foot shaft; (3) in chain elevators, to prevent loose chain from climbing up on the teeth of the foot wheel; (4) in belt elevators, to maintain a tension T_2 in the down belt by applying a load to it which will act regardless of the normal stretch in the operation of the belt.

The last item is the most important, and it is the one to be considered here. For a discussion of it, see page 365. It is evident that if the load is light and the driving conditions at the head of the elevator are favorable there is little or no need for artificial tension in the belt; but if the load is heavy and if the belt is dirty, wet, or works in an atmosphere of dust, the coefficient of belt contact falls off and then there is need of artificial tension to maintain the proper ratio of $\frac{T_1}{T_2}$. There is also need for artificial tension if the belt is heavily loaded even though clean; that condition, or an overload in the boot, has the effect of increasing T_1, and unless T_2 is great enough the belt will slow down and slip.

The great practical advantage of a weighted take-up on a belt elevator is that a steady artificial tension can be applied to the down belt which will keep T_2 high enough to prevent slip and avoid a choke in the elevator. On the effect of belt slip, see page 370.

Artificial Tension in Vertical Elevator Belts. Table 23-A has been prepared to show what artificial tension is necessary in vertical elevators for various conditions of drive and for various values of f, the coefficient of belt contact. For instance, an elevator handling a clean, sized material has a belt that weighs 2.3 pounds per foot; empty buckets weigh 4.3 pounds per foot, and the material in the buckets 3.5 pounds per foot. Then the ratio $\frac{T_1}{T_2} = \frac{2.3 + 4.3 + 3.5}{2.3 + 4.3} = 1.6$ and from Table 23-A it is evident that no belt tension need be applied at the boot for any kind of drive. It is not necessary to lag the head pulley, and an iron pulley might be expected to drive the belt even though the surfaces in contact were wet, provided that the work of picking up material from the boot were not great. If the buckets carried a heavier material, or if the pick-up were harder, the ratio $\frac{T_1}{T_2}$ might rise to 2.0 and then it would be necessary, for wet conditions, to load the down belt (and necessarily the up belt) at the boot with about 15 per cent of the dead weight of the down run. That is, if the empty belt and buckets on the down side weighed 400 pounds, then the added load at the boot (considering both runs of belt) would be $400 \times 0.15 \times 2 = 120$ pounds.

When the buckets must dig their load from the boot the pull in the up belt is more than the weight of the belt and loaded buckets by the pull required to dig. In grain elevators the buckets, and often the belt, are relatively light; this factor, on the one hand, and on the other the considerable pull required to dig, combine to raise the ratio of $\frac{T_1}{T_2}$ to 2.5 or more. In the example quoted in Chapter 22 the ratio was 2.63, and from Table 23-A it is evident that under the operating conditions—a rubber-covered head pulley working in a dust—it would be necessary to appply to the belt in the boot a load of 22.5 per cent $\times 2 = 45$ per cent of the weight of the down belt with its buckets, or $2700 \times 0.45 = 1215$ pounds.

Varying the Boot Tension with the Load. When the operating capacity of a grain elevator is based upon not more than 80 per cent of the level-full capacity of the buckets, as rated in the catalogs of manufacturers, there is generally margin enough to provide for ordinary contingencies of feed; but if the feed is irregular and subject to fluctuations beyond the normal rate of loading, or if the capacity of the

elevator is to be forced at times beyond 80 per cent of the rated capacity of the buckets, then provision should be made to apply extra belt tension at the boot, if necessary, so that the ratio of $\frac{T_1}{T_2}$ does not become too large for the driving conditions at the head of the elevator.

TABLE 23-A

ARTIFICIAL TENSION FOR VERTICAL BELT ELEVATORS

T_1 = weight of loaded belt and buckets, plus pull to dig.
T_2 = weight of empty belt and buckets.

Tension to be applied at boot to maintain driving contact at head pulley. Figures are percentages of T_2 to be added to T_1 and to T_2.

1	2	3	4	5	6	7
Ratios of $\frac{T_1}{T_2}$	Rubber-covered Pulleys			Plain Iron Pulleys		
	$\frac{T_1}{T_2} = 3.00$. Clean. $f = 0.35$	$\frac{T_1}{T_2} = 2.33$. Dusty. $f = 0.27$	$\frac{T_1}{T_2} = 1.87$. Wet. $f = 0.20$	$\frac{T_1}{T_2} = 2.19$. Clean. $f = 0.25$	$\frac{T_1}{T_2} = 1.87$. Dusty. $f = 0.20$	$\frac{T_1}{T_2} = 1.87$. Wet. $f = 0.20$
1.8	0	0	0	0	0	0
1.9	0	0	3	0	3	3
2.0	0	0	15	0	15	15
2.1	0	0	26	0	26	26
2.2	0	0	37	1	37	37
2.3	0	0	49	9	49	49
2.4	0	5	60	17	60	60
2.5	0	13	72	26	72	72
2.6	0	20	84	34	84	84
2.7	0	28	95	43	95	95
2.8	0	35	107	51	107	107
2.9	0	43	118	60	118	118
3.0	0	50	130	68	130	130
3.1	5	58	141	77	141	141
3.2	10	65	153	85	153	153
3.3	15	73	164	93	164	164
3 4	20	80	175	102	175	175
3.5	25	88	187	110	187	187

Illustration. In the elevator, the boot of which is shown in Fig. 23-21, the empty belt and buckets on the down leg weighed about 4200 pounds, and for a capacity of 20,000 bushels per hour the loaded belt on the up leg weighed about 10,000 pounds; allowing 800 pounds

for the digging and friction in the boot, the ratio of belt tensions is $\frac{10,800}{4200} = 2.57$, and from interpolation in Table 23-A, column 3, the total weight to be applied to the boot is $4200 \times 0.179 \times 2 = 1500$ pounds. For a capacity of 22,500 bushels per hour the ratio is $\frac{11,500}{4200} = 2.74$, and the weight required is $4200 \times 0.308 \times 2 = 2600$ pounds. For the maximum rating of the elevator, 25,000 bushels per hour, the ratio is $\frac{12,200}{4200} = 2.9$, and the weight required is $4200 \times 0.43 \times 2 = 3600$ pounds. These values, of course, are based on the assumed coefficients of belt contact as stated in Table 23-A, but they agree quite well with the best practice in grain-elevator design and operation.

On the permissible unit stresses in elevator belts, see page 369.

Varieties of Automatic Boot Take-ups. A simple way is to make the foot pulley heavy enough, or add weights to the boot shaft, to take up the slack or maintain the proper tension on the belt. In some belt elevators with 36-inch 10-ply belts, used by a western copper-mining company, the foot pulleys are 42 inches in diameter with rims 2½ inches thick. The foot bearings slide in guides, and no take-up screws are used. An old device is a pair of levers, one on each side of the boot, pivoted at one end, weighted at the other end and with an intermediate link bearing down on top of the boot bearing. One arrangement is shown in Fig. 23-24. Several patents of no worth have been issued on arrangements where the whole boot is hanging on the loop of belt and is guided by sleeves or tracks on the lower part of the elevator casing. A few elevators have been built where the whole weight of the boot is carried on a pivoted arm so as to maintain a tension on the elevator belt or chain, and since the wheel does not travel within the boot, a fixed distance is preserved between the sweep of the buckets and the bottom of the boot. This is an advantage in handling lumpy materials and materials like fertilizers, which form a hard, dense crust

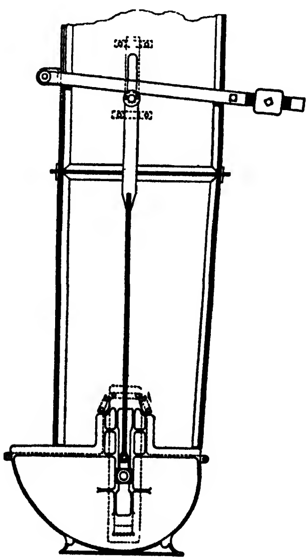


FIG. 23-24. Weighted Lever Device for Automatic Take-up.

beyond the sweep of the buckets, but it is of no value in handling grain or similar substances (see Fig. 23-8).

Edmond Take-up. A practical device used on many large grain elevators is the Edmond automatic take-up (Edmond-Norell patent of 1911). It consists of a weighted frame, like that shown in Fig. 23-21, for maintaining the tension on the belt, plus means for independently moving the foot bearings in the frame so as to adjust the level of the shaft and cause the belt to run true on the foot pulley. In practice, the lead of the down belt on to the foot pulley is also controlled by adjusting the deflector pulley which in large elevators is placed in the down leg where it leaves the vertical to slant to the boot. A very slight deviation of this pulley from true level will train the belt one way or the other on the foot pulley.

Fig. 23-19 shows a belt elevator boot in which the belt tension is maintained by a weighted lever device.

Chokes in grain-elevator boots are due to various causes:

1. Elevating capacity of the buckets is too small for the feed.
2. Bins become full; grain backs up in head chute, then falls down back leg.
3. The supply of current to the motor is interrupted and the elevator stops suddenly; or the voltage falls, and the motor fails to pull the load; or there is a slip or failure in the power transmission.
4. Sticks, tools, etc., get into the boot, foul the belt, and prevent it from moving.
5. Upper end of discharge chute is set too high for a clean discharge under all conditions.
6. Strings or pieces of bagging catch on the upper end of the chute, prevent a clean discharge, and cause spill down the back leg.
7. If the elevator is shut down while loaded and is not fitted with a backstop, the elevator may run backward and fill up the boot and lower part of the casing.
8. Speed is too high; grain hits the top or front of the hood and falls down the back leg.
9. The elevator is stopped before the supply of grain to the boot is shut off.
10. Take-up tension is not maintained. (On this point, see page 365.)

When the choke is such that the belt slows down or stops while the power is still on and the head pulley continues to turn, the result is that the pulley rubs hard on the inside of the belt, wears the plies

away, and often generates heat enough to start a fire and cause a dust explosion. A report on the cause of a disastrous explosion says:

> There was no question after investigation but that a choke had occurred in an elevator leg at ten minutes to twelve. The men left the plant shortly after, and on their return at noon smelled the odor of burning rubber, which they thought was due to a hot belt in the basement, a belt which had been rubbing to one side. We found a man who had been to the top floor of the elevator and only about two minutes before the explosion saw smoke coming out of the elevator, and saw the flames of the burning belt, and only had time to get to the first floor before the explosion. . . . There was no general fire; when men entered the plant less than half an hour after the explosion, that particular elevator leg was red from fire; it was the only one in which there was fire. The belt was burned in two.*

In many chokes the belt slips but does not stop; in this case the pulley side may be seriously torn and frayed by the continued rotation of the head pulley, especially when the lagging or covering of the pulley is in bad condition, with bolts projecting (see page 342). The heating of the belt due to frequent slipping of this kind may, in combination with excessive creep due to high belt tension (see page 370), tend to "age" the friction in rubber belts or cause canvas belts to dry out and crack. This is more true of oil-saturated belts (class 1 impregnation) than of belts saturated with asphaltic compounds, which resist heat better and are not affected to the same degree.

Foot Pulleys Too Small. It is certain that much of the prevalent trouble with choked grain boots is due to the use of foot pulleys that are too small. When a boot pulley revolves three or four times as fast as the head pulley the forces which prevent the grain from staying in the buckets under the foot wheel are five to seven times as great as those which throw the grain out of the buckets on the head wheel (see page 298). This prevents the buckets from taking any load until they are on the point of leaving the foot wheel or are on the straight lift. All the while they are in contact with the foot pulley they merely stir up the grain, add to the pull on belt and bolts, and consume power. When the pulley is larger, the conditions for pick-up are better (compare Figs. 17-1 and 17-3), the boot is larger, and there is more room in it to take care of an accidental accumulation of grain; the excess of feed over elevator capacity, or what spills down the back leg, will not pile so deep on the rising side; the work of digging will be easier and chokes will not be so frequent or so harmful.

* Dr. H. H. Brown, U. S. Dept. of Agriculture, reporting to the U. S. Grain Corporation on the cause of an explosion in a grain elevator at Port Colborne, Ontario, in which ten men were killed. (*Proceedings* of Conference on Grain Dust Explosions, April 24, 1920.)

Lack of Automatic Take-up. For reasons stated on pages 365 and 372, grain-elevator belts will slip if the load is heavy and if the belt tension is not sufficient. As between no load and full load, a belt may stretch some inches in 100 feet; if an automatic take-up is not used, and if screws are used to tighten the belt, neglect to adjust the screws during the operation of the elevator may cause the ratio of belt tensions $\frac{T_1}{T_2}$ to fall to such a point that the head pulley will not drive the belt. Then the belt slips and a choke occurs. The same thing may happen if the weight on an automatic take-up is not sufficient; but in any case, an automatic take-up is better than a screw take-up on any grain elevator; in high elevators it is really essential.

Prevention of Chokes in Grain Elevators. In the design of the elevator the bucket capacity should be larger than is necessary to give the required number of bushels per hour; for reasons stated on page 324 it should be great enough to cover the peak-load capacity which, for a minute or a few minutes, may be at a rate sufficient to exceed the hourly rate by a large margin. An excess of bucket capacity is a kind of insurance; it takes care of emergencies, and if a choke does occur, the elevator can dig itself clear in a short time without damage to the belt or buckets.

The lip of the head chute should come close to the buckets and it should be set not less than 15° below the center of the head shaft if the belt travels at the speeds of Table 17-A; 20° is better. If the belt speed is lower, Table 17-B, the angle should be 30° for grain. Many high-speed elevators have chutes set 12° or 15° below the head shaft; but there is always the risk that the speed may fall off, or that, for some other reason, the buckets may not discharge properly into a chute set too high. To set the lip of the chute at 20° instead of 12° means an increase of only 7 or 8 inches in the height of an elevator with a 72-inch head pulley—not a heavy price to pay for some insurance against trouble.

There should be plenty of room in the front of the hood over the head pulley, so that the direct discharge and also the splash from the buckets will enter the chute without striking the front plate.

Fig. 23-25 shows the head of a large grain elevator with the lip placed lower than has been customary. The angle from the center of the head shaft down to the upper edge of the rubber belt which forms the lip (see detail) is 25°. All the joints at and above the level of the head shaft are bolted, as is the sloping plate which joins the two legs below the head pulley. The figure shows the bar screen referred to on page 354 and the door which gives access to it.

A number of safety devices are in use to prevent grain from dropping down the back leg in case the flow from the discharge spout is stopped by the chute filling up or by a choke in the spout itself. An auxiliary by-pass spout is fitted to the discharge spout near the head casing so that, when the main spout fills up, the grain will enter the by-pass and be directed to an emergency bin, or be dropped on a floor where it can be seen, or made to enter a counterbalanced tank, which, on receiving a certain weight of grain, drops, sounds an alarm, and

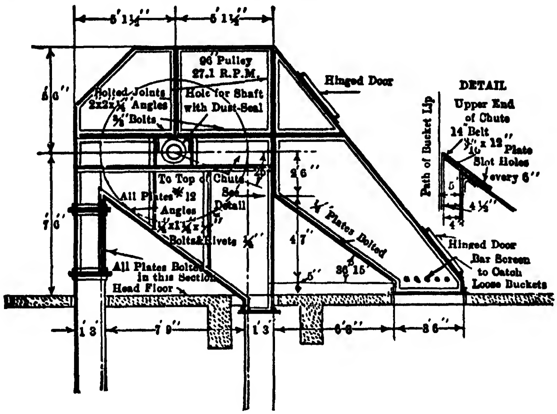


FIG. 23–25. Head of Shipping Elevator, Public Grain Elevator, New Orleans. (Ford, Bacon & Davis, Engineers.)

closes the gate at the boot or else stops the conveyor which delivers to the boot.

On the influence of automatic take-ups and large foot pulleys in reducing the risk of chokes in the boot, see above.

When a choke does occur it is important that the rotation of the head pulley should stop as soon as the belt slows down; otherwise the belt may be damaged or burnt. When the drive is from a separate electric motor the danger of a burnt belt is reduced when the circuit breaker has an overload release set to throw out before the overload becomes too great. Electrical safety devices are used also to stop motors when bins are full, to stop the conveyor which delivers to the boot as soon as current is cut off from the elevator motor, and to prevent the conveyor motor from starting until the elevator is up to full speed.

The Hall non-chokable boot is a cast-iron boot with the entrance of the chute opposite the low position of the shaft. A baffle or barrier, *A*, Fig. 23-26, limits the flow of grain to the boot, and the parts are arranged so that, if the buckets are overloaded and push up past the

opening of the chute more grain than they can carry, the surplus spills over the baffle into the chute. This spilled grain forms a surcharge to the grain as it lies in the chute at its natural angle of repose, and it must be delivered to the boot before the flow of grain in the chute can start again. The excess amount of grain acts, therefore, as a check to prevent choking in the boot. The view on the left shows the condition when the elevator is at rest; the one on the right shows the action when the belt is running.

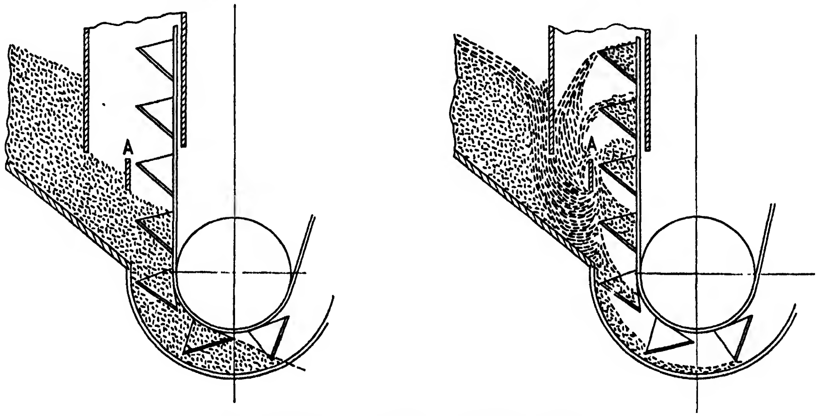


FIG. 23–26. Hall Non-chokable Boot.

Elevator Back Stops. To prevent the elevator from running backward under load with the power shut off there may be on the end of the head shaft a ratchet wheel which engages a pawl fulcrumed on the beam that supports one of the shaft bearings. To avoid noise, the pawl is usually fitted with a friction device of some kind which keeps it away from the ratchet teeth while the shaft revolves in the operating direction, but allows it to drop into place and stop the ratchet wheel if the shaft should start to turn backward.

Another device is a stop applied to a rope sheave on an elevator head shaft. A hardened-steel roller is contained in a frame on which is mounted a brake shoe, engaging one or two of the grooves on the lower or empty side of the sheave. The roller bears against a finished part on the inside of the sheave rim and is backed up by a wedge-shaped steel plate. In normal rotation, the roller stays at the small end of the wedge plate, and the brake block hangs clear, but if the sheave starts to reverse, the roller travels toward the thick end of the wedge plate and brings the brake block into contact with the grooves of the sheave.

When an elevator is driven by its own electric motor, a solenoid brake is a good backstop.

CHAPTER 24

INCLINED ELEVATORS

Inclined Elevators. For the same belt speed and same size of foot wheel the pick-up at the foot of an inclined centrifugal discharge elevator is apt to be better than in a vertical elevator, because, at the moment of leaving the wheel to enter on the straight inclined run, the bucket is tilted back and the resultant of pressure due to the combination of centrifugal force and gravity is directed more toward the bottom of the bucket and is not so likely to force material out over the front lip. This is not so noticeable when the mass of material in the boot is small and when the buckets do not fill; but when the mass is deeper, and the loading continues until the lip of the bucket is at or above the level of the foot shaft, then the last material delivered to the bucket is jerked toward the back when the bucket enters on the straight inclined run and does not partly spill over the front lip, as happens when the run is vertical.

Fig. 24-1 illustrates this difference by showing that for speeds and sizes of wheels as shown in Fig. 17-5, the resultant pressure for the bucket on a 20° incline makes an angle of 32° with the back of the bucket, while the pressure for a bucket on a vertical run is 15 per cent greater in amount and inclined at 44° to the back.

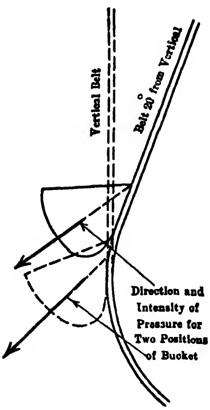


Fig. 24-1. Conditions for Filling Buckets at Foot Wheel More Favorable When Elevator Is Inclined.

After passing the foot wheel, the belt and buckets vibrate to some degree, and the material, if free-flowing, tends to shake down to a surface at

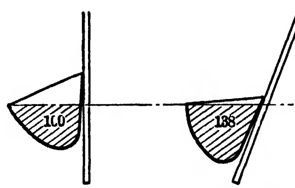


Fig. 24-2. Comparison of Loading Malleable-iron *A* Buckets on Vertical Run or 20° Incline.

right angles to the direction of gravity—that is, the surface of the material in the bucket will be approximately horizontal. This means that a bucket piled full on leaving the foot wheel is likey to spill less

on an inclined run than on a vertical run. Fig. 24-2 shows that the water-level capacity of a standard malleable-iron Style A bucket is 38 per cent more on a 20° incline than on a vertical run. Though this is not true to the same extent of materials which do not flow freely and which pile up at steep angles, still it is generally true that standard Style A buckets carry better on inclines up to 30° from the vertical than on the vertical.

When the bucket reaches the head wheel the material in it comes under the action of centrifugal force, and if the speed is high enough, and if the material lies up to the lip of the bucket, some of it may spill over the lip. Whether this happens or not depends on the shape of the bucket and the direction of the resultant pressure at the point where the

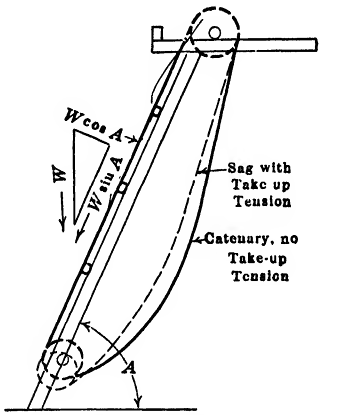


FIG. 24-3. Hang of Return Belt on Inclined Elevator as Affected by Take-up Tension.

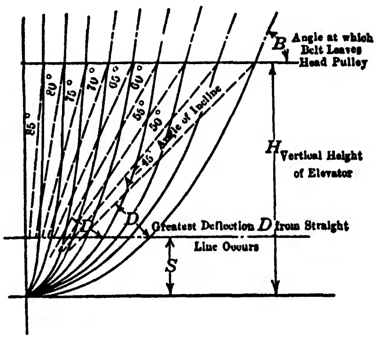


FIG. 24-4. Forms of Catenary Curves for Return Belt for Different Angles of Slope.

belt meets the wheel. Reference to Figs. 17-1 and 17-4 will show that the pressure at position 4 in ordinary centrifugal discharge elevators is less likely to cause spill than the pressure at position 3, because it is less in intensity and is directed more toward the bottom of the bucket and not so much toward the lip. The same thing is true of any position on the wheel between 3 and 4 where an inclined belt meets it; hence in an inclined elevator a bucket loaded to the edge of the lip is less likely to spill on reaching the head wheel than the same bucket in a vertical elevator with the same size head wheel and same belt speed.

Path of Belt on Inclined Elevators. Fig. 24-3 represents an inclined belt elevator with the up run supported on idlers and the down run hanging free. If no tension is applied to the belt at the foot wheel, the down run forms a catenary. Fig. 24-4 shows the forms of

catenary curves for elevators inclined from 45° to 85° from the horizontal, and Table 24-A shows characteristics of these curves. The use of the table can be understood from an example taken from practice. In a heavy stone elevator inclined 65° from the horizontal,

TABLE 24-A

FACTORS FOR INCLINED BELT ELEVATORS WITH UP RUN SUPPORTED AND RETURN RUN HANGING FREE (NO ADDED TAKE-UP TENSION)

1	2	3	4	5	6	7	8	9	10
Angle A	L = Factor for Length of Down Belt	K = Factor for Tension in Down Belt	S = Factor for Locating Maximum Sag	D = Factor for Deflection of Down Belt	Angle B	Angle of Belt Wrap	Ratio $\frac{T_1}{T_2}$ for f = 0.25	Ratio $\frac{T_1}{T_2}$ for f = 0.35	Sin $A + f$ cos A = Factor for W
45	1.50H	1.62wH	0.26H	0 20H	67°34′	157°26′	1.99	2.62	0.78W
50	1.39H	1.46wH	0 26H	0 19H	71°36′	158°24′	2.00	2.64	0.82W
55	1.30H	1.34wH	0.26H	0 18H	75°9′	159°51′	2 01	2.66	0.85W
60	1.23H	1.25wH	0.25H	0 16H	78°23′	161°37′	2.03	2.69	0.89W
65	1.17H	1.18wH	0.25H	0.14H	81°7′	163°53′	2.05	2.71	0.94W
70	1 12H	1.13wH	0.24H	0.12H	83°32′	166°29′	2.07	2.77	0.96W
75	1.08H	1.08wH	0 23H	0.10H	85°39′	170°29′	2.10	2.83	0 98W
80	1.05H	1.05wH	0 22H	0.07H	87°28′	172°33′	2 13	2.87	0.99W
85	1 02H	1 02wH	0 20H	0 04H	88°57′	176°03′	2 16	2 93	1 00W
90	1.00H	1.00wH	......	. ...	90°	180°	2 19	3.00	1 00W

w = weight of empty belt and buckets per foot (pounds).
W = total weight of belt buckets and material on loaded run (pounds)
H = vertical height in feet.

the belt was 38-inch 10-ply and weighed 12 pounds per foot; steel buckets empty weighed 50 pounds per foot; the height on the incline was 67 feet, and the vertical height 60 feet.

From column 2, the length of belt from where it leaves the head pulley to where it meets the foot pulley is $1.17 \times 60 = 70.2$ feet.

From column 3, the pull in the return belt at the top is $1.18 \times 60 \times (50 + 12) = 4390$ pounds.

From column 4, the greatest sag occurs at a point in the belt $0.25 \times 60 = 15$ feet vertically above the bottom of the foot pulley.

From column 5, the deflection at that point from a straight line touching the two pulleys is $0.14 \times 60 = 8.4$ feet (measured at 90° to the straight line).

Column 6 shows that the return belt is inclined at 81° from the horizontal (9° from vertical) where it leaves the head pulley, and from this we find that the angle of belt wrap on the head pulley is about 164° (see column 7).

When, by means of take-ups or adjustable bearings, tension is applied to the belt, the curve changes its shape, becoming more like a parabola; the deflection D becomes less, and the point of maximum sag rises slightly. In practice it is generally necessary to put tension in the return belt to stop it from swaying too much and also to increase the driving effect at the head pulley by making T_2 larger. When several stone elevators inclined 15° to 25° from the vertical were measured under operating conditions, the actual deflection of the return belt was found to be ½ to ⅔ of D. Nevertheless, the catenary should always be laid out to find the length of the belt, to locate the lip of the head chute, and to show what clearance is required back of the elevator in case the belt should run slack.

Calculation of Applied Tension. If in Fig. 24-3 W represents the total weight of the loaded up run, belt, buckets, and material carried, then the downward component of W which puts a direct pull in the belt is W sine A, and the component which makes the idlers turn is W cosine A; if f is a coefficient which includes the friction of the belt idlers and the slight lifting of the load in passing over the idlers, then fW cosine A is the belt pull due to contact with the idlers. The total belt pull due to the weight is, therefore,

$$\text{Pull} = W(\text{sine } A + f \text{ cosine } A)$$

To this should be added the pull due to pick-up (see page 367).

Column 10, Table 24-A, gives values of (sine $A + f$ cosine A) in which f varies from 0.05 on steep elevators to 0.10 for an incline of 45°, the higher coefficients allowing for the greater motion of the belt and load in passing over idlers where the incline is not steep. The figures in column 10 assume that the material is fed to the buckets from a chute and is not dug from a boot.

In large belt elevators handling stone and ore in continuous buckets the weight of material carried is great in proportion to the weight of the empty belt and buckets; and for reasons already mentioned in Chapter 22 it is necessary to tighten the belt by means of the take-ups. In the stone elevator referred to above the buckets held at full capacity 130 pounds of stone per linear foot. The length of the inclined loaded run is 67 feet; its weight is therefore $67(12 + 50 + 130) = 12{,}864$ pounds, and from column 10, Table 24-A, the pull at the head due to

the weight is $0.94 \times 1{,}2864 = 12{,}093$ pounds. For the pick-up of material we may allow the equivalent of 11 feet on the lift (see page 367) or 1430 pounds, making a total pull of $T_1 = 13{,}523$ pounds.

Since the pull at the top of the return belt is 4390 pounds, the ratio of $\frac{T_1}{T_2}$ for a natural sag of the return belt without applied tension is $\frac{13{,}523}{4390} = 3.08$. This is greater than 2.71, which is the working ratio of $\frac{T_1}{T_2}$ when $f' = 0.35$ (see column 9); hence not even a lagged pulley will drive the elevator unless some take-up tension is added. To find out what tension x is necessary, assume that the head pulley is lagged and that the ratio of $\frac{T_1}{T_2}$ is to be 2.7, then

$$13{,}523 + x = 2.7(4390 + x)$$

from which $x = 980$ pounds.

When the take-up puts 980 pounds tension into each run of the elevator the pull in the up belt is $13{,}523 + 980 = 14{,}503$ pounds and the 36-ounce duck is stressed to $\frac{14{,}503}{38 \times 10} = 38$ pounds per inch per ply.

Discharge from Inclined Elevators with Spaced Buckets. In a vertical elevator some clearance space must be left between the upper edge of the discharge chute at the head and the lips of the descending buckets. Although the space may be small, some drip from the buckets, or material delayed in discharge, may fall through it and be wasted; but if the elevator is inclined, the chute may be placed partly under the path of bucket travel so that some or all of the spill may be caught.

This point is not of much importance in handling free-flowing materials like grain. These can be picked up and discharged at high speed, but if the material is damp and sluggish, or if it is hard, lumpy and abrasive, lower speeds like those of Table 17-B are preferable; or it may be advisable to pick up and discharge difficult material at speeds still lower. If the elevator is run at speeds lower than Table 17-B it *must* be inclined in order to catch the spill and the delayed discharge. This condition of discharge is illustrated in Fig. 17-13 and it is shown also in Fig. 24-5. If Fig. 24-5 is turned so that the line *A—B* is vertical, the view represents an elevator inclined at 40° from the vertical and run at a speed less than Table 17-B. A chute at C will catch the material which begins to fall from the buckets high on the wheel, and also some of what is scattered by interference between bucket D and the discharge from bucket E.

Size of Head Wheels. The bad discharge caused by the interference mentioned above causes a waste of material and power, because even in an inclined elevator a chute set partly under the head will not catch all the scattered material. It is evident from Fig. 24-5 that if the head wheel were 18 inches in diameter instead of 36 inches the buckets which

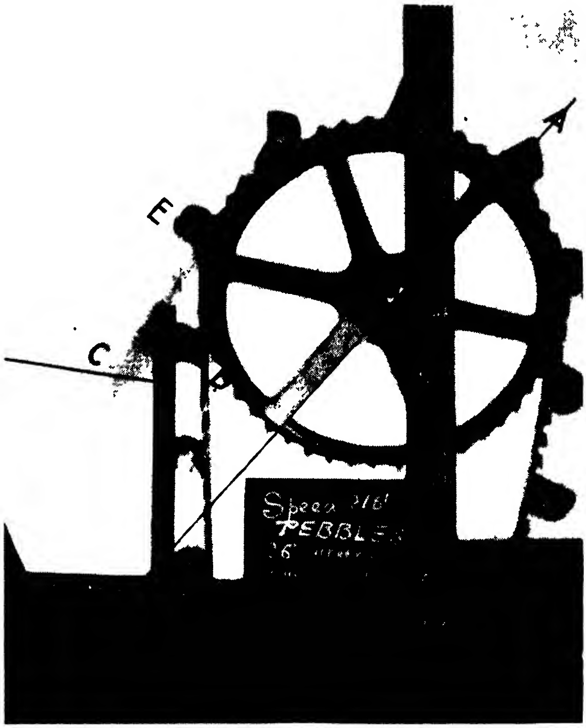


FIG. 24-5. Condition of Discharge for Which It Is Necessary to Incline the Elevator.

come every 12 inches would be 76° apart on the wheel instead of 38° apart as shown in the figure. With this greater angle between the adjacent buckets, the discharge would be cleaner and interference would not occur.

This is shown better in Fig. 24-6, representing the head of an elevator inclined 30° from the vertical. With the large wheel, the positions of consecutive buckets are 1, 2, 3, etc. With the same linear spacing of buckets on a head wheel of half the size the corresponding positions are 1, 4, 5, etc. If the speed is such that discharge begins at 1, and follows the parabola 1—*G*, some of it will hit the bucket at 2; but

such a discharge would clear the bucket at 4 just leaving the small head wheel.

Discharge with Small Head Wheel. As a matter of fact, the discharge from a bucket at 1 on the small wheel does not follow the path 1—G, but goes out on the parabola 1—P, because, for a given belt speed, centrifugal force at a head wheel varies inversely as R—that is, it is twice as great on a wheel of half the size, *for a given belt speed.* The two arrows drawn from 1 represent by their position and their length, as drawn, the direction and intensity of the resultant forces which cause material to leave the bucket at 1, and the parabolas are tangent to them.

This explains why, in an inclined elevator, run at a given speed in feet per minute, a small head wheel makes a better discharge:

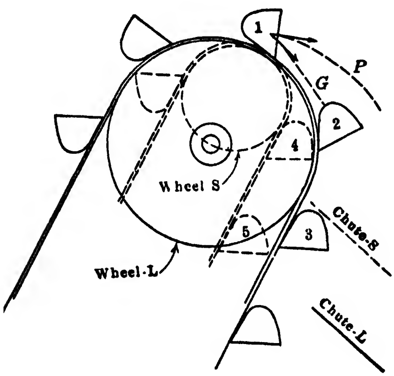


FIG. 24-6. Effect of Wheel Diameter on Discharge at Head of Inclined Elevator.

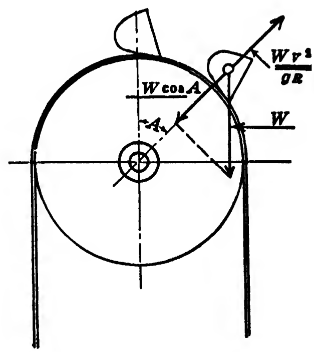


FIG. 24-7. Point at Which Discharge Begins.

1. The buckets turn more quickly under the wheel and get out of the way of discharged material.

2. Centrifugal force is more active and there is more "throw" to the discharge.

Point at Which Discharge Begins. In considering the action at the head of an inclined elevator run at comparatively slow speed, the point at which discharge begins may be referred to. According to theory, the material begins to leave the bucket on the descending side of the wheel at the point at which centrifugal force equals the radial component of the weight. In Fig. 24-7 the forces which determine discharge are (1) gravity, acting downward with a force W, or if measured on the radial line, W cosine A, where A is the angle from the vertical; (2) centrifugal force, acting radially outward with a value

$\frac{Wv^2}{gR}$ (see page 297). The mass within the bucket will be in equilibrium, tending neither to fall out nor fly out when $\frac{Wv^2}{gR} = W$ cosine A or when cosine $A = \frac{v^2}{gk}$.

When $v^2 = gR$, as at the high speeds of Table 17-A (see page 297), cosine $A = 1$, $A = 0°$, and discharge is ready to begin at the top of the wheel; when $v^2 = \frac{2}{3}gR$, as at the lower speeds of Table 17-B (see page 302), cosine $A = \frac{2}{3}$, $A = 48°$, and discharge may be said to begin about halfway down in the quadrant of discharge. It is a fact, however, that some material always leaves the bucket sooner than would be indicated by the angle A calculated in this way. It is caused by the shifting of the load within the bucket; at position 4, Fig. 17-4, the resultant pressure is directed toward the bottom of the bucket; at 6 it is directed toward the back; since the surface of the material tends to arrange itself at right angles to the line of pressure, the load shifts in the bucket; at 6 it flattens out on the back of the bucket, and if the bucket is full or nearly full some of the material may spill out over the leading edge of the back. This can be counteracted by making buckets with high backs or hooded backs, but ordinary buckets on belt run at slow speed are always likely to spill if they carry a full load.

Spill Due to Shift of Load within the Bucket. This spill is not so likely to occur in high-speed elevators, because discharge occurs before the load has time to shift in the bucket, and if it does occur, it may be affected by centrifugal force to a degree sufficient to describe a parabola that will land it in the head chute; but in elevators run at speeds less than Table 17-B, any material spilled between the top of the wheel and the point of theoretical discharge will fall so nearly vertically that the only way to catch it is to incline the elevator and set the chute well under the head wheel. The loss of some of this material is unavoidable, being due to interference with the descending buckets, but some material will enter the chute.

Size of Head Wheels and Spacing of Buckets. When, for the sake of slow speed in pick-up or discharge, inclined elevators are run at speeds one-half of those in Table 17-A, the effect of centrifugal force becomes very small and the main discharge from a bucket occurs more than halfway down in the discharge quadrant—that is, angle A, Fig. 24-7, is over 70° from the vertical. The path of discharge is then nearly vertical, and in order to avoid interference the leading bucket should be so far ahead on the arc of travel that the discharged material cannot fall on it. The spacing of buckets in slow-speed inclined elevators can

therefore best be expressed as an angular distance on the circumference of the head wheel; and the diameter of the head wheel should be such that a normal bucket spacing should not cover too small an arc. In other words, the head wheel must not be too large.

When elevators inclined at about 30° from the vertical are run at speeds one-half those of Table 17-A or less buckets of standard size and shape make a satisfactory discharge if they are spaced 90° apart on the head wheel—that is, if the circumference of the head wheel is four times the bucket spacing.

Table 24-B shows wheel sizes and revolutions and bucket spacings for belt speeds one-half those of Table 17-A. It gives also the projections of buckets proper for the given spacings. In elevators designed in this way centrifugal force equals one-fourth the force of gravity, and the angle A, Fig. 24-7, at which the main discharge occurs, is 75½° (cosine = ¼).

At speeds lower than those of Table 24-7 the influence of centrifugal force is practically zero, but the data given still hold good as to wheel sizes, bucket spacings, and bucket projections.

Advantages of Inclined Elevators. Inclined elevators are often preferred to vertical elevators for reasons other than those of pick-up and discharge at slower speed. If there is a horizontal traverse from

TABLE 24-B

HEAD WHEELS, SPEED, BUCKET SPACING, INCLINED ELEVATOR, CENTRIFUGAL DISCHARGE AT LOW SPEED

Diameter of Wheel, inches	Revolutions per minute	Belt Speed, feet per minute	Bucket Projection, inches	Bucket Spacing, inches	Diameter of Wheel, inches	Revolutions per minute	Belt Speed, feet per minute	Bucket Projection, inches	Bucket Spacing, inches
12	35	110	3	9 4	27	24	175	7	21.2
15	31	124	3½	11 8	30	23	180	8	23 5
18	28	132	4	14.1	33	22	190	9	25.9
21	27	147	5	16.5	36	21	198	10	28.2
24	25	157	6	18 8					

the loading point to the discharge point of not more than one-half the height through which the material is to be lifted, setting the elevator on an incline will both elevate and convey the material without the use of a feed conveyor at the foot or a bearing-off conveyor at the head. When, however, the angle of incline is more than 25° from the vertical,

it becomes necessary to support the loaded run and to provide space for the sag of the return run (see Table 24-A).

One of the merits of the inclined elevator shown in Fig. 23-7 is that the fine damp sand which discharges late, or which shakes off the buckets at the head, does not fall down on the foot pulley as it would if the elevator were vertical. The vertical back wall of the casing permits the spill to fall straight down into a clearance space in back of the belt, from which it is picked up by the buckets when the accumulation becomes large enough. Keeping the sand from falling onto the pulley lessens the injury to the pulley side of the belt due to sand and grit between it and the pulley.

When a vertical elevator with centrifugal discharge is shut down in an emergency while loaded, the speed falls off, the buckets make a bad discharge, and the material misses the chute and falls down into the boot, with the possibility that the boot may be choked when the elevator starts again. In an inclined elevator with the chute partly under the head wheel most of the spill will be caught and will not choke the boot.

In handling mineral pulps and slimes an elevator slightly off the vertical is preferable for the reasons stated on page 404: the spill in passing from the foot wheel to the straight run is less; the buckets hold more on the incline; the spill is less on meeting the head wheel; and if the incline is enough to permit the head chute or receiver to be placed partly under the path of the descending buckets some of the solids which are discharged late will be caught instead of spilling down into the casing. The inclination is generally limited to 10° or 15° from the vertical; otherwise, the up run must be carried on idlers, and they are troublesome in a wet elevator.

A general advantage of a belt elevator inclined at 20° or more from the vertical is that the sag between head and foot keeps the belt in contact with the foot pulley in spite of occasional neglect of the take-ups. If the foot pulley in a vertical elevator is not set down to follow the stretch of the belt the contact between them may be so slight that the pulley and shaft will not turn, and then the belt will rub and wear. If the slack is so great that the belt is loose on the pulley the buckets are not backed up by the pulley; they do not fill well, and the capacity of the elevator falls off.

The length of the return belt is also greater than in a vertical elevator of the same height; and when the take-ups are set down to apply added driving tension to the belt, the belt can work longer and stretch more before the effect of the take-up tension is lost.

This may be expressed in a different way by saying that if the

inner curve in Fig. 24-3 represents the hang of the belt when pulled to its maximum operating tension, and if the catenary shows the hang of the belt under no take-up tension, then the difference in length between the two is the amount the belt may stretch in service before it loses all the effect of the take-up tension. This difference may be a foot or more in an inclined elevator, but the corresponding difference in a vertical elevator may be only a few inches. When the vertical belt stretches a few inches in service it may slip on the head pulley and act badly at the foot unless the take-ups are set down; but an inclined belt may stretch several times as much before it is necessary to adjust the take-ups again.

For the reasons just stated, inclined belts do not have to be shortened and respliced so frequently as vertical belts, and a new inclined belt will run longer before it has to be cut and shortened.

In stone and rock elevators inclined at 20° or 25° from the vertical the sag of belt acts to some extent as a relief if a lump catches between the belt and the foot pulley. In a vertical elevator a tight belt is not likely to stretch still further and prevent injury when an accident of this kind occurs. In an inclined elevator, however, with some free sag at the bottom of the down belt, there may be "give" enough to prevent the stone from punching a hole in the belt. This is merely an incidental reason for inclining a stone elevator with continuous buckets on a belt; the main reason is that the incline permits the buckets to be loaded from a chute without spill (see page 330).

Disadvantages of Inclined Elevators. When an inclined elevator is high, or is set at a considerable angle from the vertical, it is necessary to support the loaded run to keep it from flapping and spilling material from the buckets. The idlers used for this purpose are flat-face pulleys like those made for the return run of belt conveyors; sometimes in heavy elevators with continuous buckets they are pipe rolls set every 6, 8, or 10 feet. Bearings for idlers on an inclined elevator are often so located that it is hard to inspect and oil them, and they are likely to suffer from neglect. Light elevators with spaced buckets can be put up without idlers if the slope is not far off the vertical, but heavy belts with continuous buckets generally require them.

A drawback to the use of inclined elevators is the greater amount of floor space required and the larger and more expensive casing necessary to enclose them.

NOTE. For a general treatment of the subject of conveying machinery see S. J. Koshkin's *Modern Materials Handling*, John Wiley & Sons, 1932, and W. G. Hudson's *Mechanical Handling Equipment*, McGraw-Hill Book Company, 1942.

APPENDIX I

WEIGHTS OF MATERIALS

Table A

Based on Circular 10, U. S. Bureau of Standards, Edition of 1924. Legal weights in pounds per *U. S. standard bushel* in various states of the United States for commodities bought and sold by measure.

Grains	
Barley	48
Bran	20
Buckwheat	48
Corn (maize), shelled	56
dry, on the ear	70
green, unhusked	40
Cornmeal	50
Malt	34–38
Oats	32
Rice	45
Rye	56
Rye flour	50
Wheat	60
Wheat flour	50

Seeds	
Alfalfa	60
Beans, dry, shelled	60
castor	46
soy	58
Clover	60
Cotton seed	30–35
Sea Island	44
Flaxseed (linseed)	56
Grass seed, blue-grass	14
red-top	14
Hungarian	50
orchard	14
timothy	45
Hempseed	44
Millet	50
Peas dried	60
Rapeseed	50
Sorghum	50

Fruits and Vegetables	
Apples, fresh	48
dried	24
Beets	56
Cabbage	60
Carrots	50
Chestnuts	50
Cranberries	33
Onions	57
Potatoes, white	60
sweet	50
Peanuts	22
Tomatoes	56
Turnips	55

Miscellaneous	
Charcoal	20
Coal, anthracite *	75
bituminous *	75–80
Coke	40
Hair, plastering	8
Lime, unslacked	70–80
Salt, Michigan	56
sea	70
fine table	60
Sand, in Pennsylvania	100
other states	130

* For greater accuracy of weights, see Table B.

1 U. S. standard bushel equals 2150.42 cubic inches, nearly 1 25 cubic feet.
1 British imperial bushel equals 2219.36 cubic inches, nearly 1.03 U. S. bushels.

Technical Paper 184, Edition of 1918, Bureau of Mines, gives weights per cubic foot of 177 kinds of coal from all parts of the United States and some foreign countries. The weights vary considerably, even in samples from adjacent mines. A fair summary of weights of anthracite and bituminous coal is given in Table B. The paper points out that the weight of coal depends not only on its percentage of ash and volatile matter but also on the quantity of surface moisture, the proportions of large and small pieces, and the amount of shaking or settling. Tests quoted show that bituminous slack increased in weight as much as 3 pounds per cubic foot when wet, bituminous nut increased in weight over 1½ pounds per cubic foot when wet, and slack when shaken diminished in volume from 6 to 8 per cent. Coal consisting of pieces of uniform size contains about 45 per cent of voids; hence mixtures like run-of-mine bituminous weigh as much as 10 per cent more than sized coal from the same mine.

Table B

Weights in Pounds per Cubic Foot of Minerals, Ores, Chemicals

Data from various sources

Acid phosphate, fertilizer	60
Alum, lump	50–60
pulverized	45–50
Ammonium sulphate, wet	80
dry	70
Asbestos, shredded	20–25
Ashes, boiler-house, dry	43
gas-producer, wet	78
Asphalt, binder for paving	80–85
Baking powder	50
Barytes, *see* Limestone, add 60%	
Basalt, *see* Limestone, add 15%	
Batch, glass plants	85–90
Bauxite, crushed	75–85
Borax, $\frac{2}{3}$ as heavy as limestone	
Calcium carbide dust	70
Cement, portland, loose	75–85
bag, 7′ × 15″ × 24″	94
barrel $19\frac{1}{2}''$ × 18″ × 29″	376
clinker	80–95
slurry	90
Chalk, *see* Limestone, less 10%	
Clay, dry, loose	63
brick, ground fine	90–110
Clinker, cement	80–95
Coal, anthracite, solid	94
large sizes	52–58
domestic sizes	52–56
steam sizes	50–54
bituminous, solid	84
run-of-mine	45–55
domestic lump	43–52
slack	43–50
pulverized for coking	30–37
lignite, broken	45–55
Coke, run-of-oven	25–30
breeze	24–35
Concrete, in place, stone	130–150
in place, cinder	110
wet, on conveyor	110–150
Copper ores, crushed	130–150
Cryolite, *see* Limestone, add 10%	
Cullet, glass plants	80–100
Culm, coal refuse	45–50
Dolomite, *see* Limestone	
Earth, as excavated, dry	70–80
wet, mud	100–110
Epsom salts	70
Feldspar, *see* Limestone	
Flue dust, blast furnace	115
wet	150
Fluorspar, *see* Limestone, add 20%	
Foundry sand, loose	80–90
rammed	100–110
Foundry refuse, old sand, cores, etc.	60–80
Fuller's earth, dry	30–35
oily	60–65
Glass, broken, *see* Cullet	
Gneiss, *see* Limestone	
Granite, *see* Limestone	
Gravel, dry	90–100
wet	100–120
Gravel and sand mixed, wet	100–130
Gypsum, *see* Limestone, less 15%	
Iron ore, depends on percentage of iron	130–180
Iron pyrites, *see* Limestone, add 50%	
Iron sulphate, pickling tank, wet	80
dry	75
Kieselguhr, infusorial earth	10–15
Lead ores, depends on percentage of lead	200–270
Lime, hydrated, pulverized	35–40
unslaked lump	60–65
Limestone, solid	165
2″ to 3″ lumps	90–95
$1\frac{1}{2}''$ to 2″ lumps	85–90
$\frac{1}{2}''$ screenings	80–90
dust	75–80
Litharge, fumed	60
skims	360
Magnesium sulphate	70
Manganese oxide	120
Marble, *see* Limestone	
Marl as dug	79
Mica, *see* Limestone	
Mortar (lime), wet	150

TABLE B—*Continued*

Peat, solid, dry	30
loose, dry	20
Pebbles, *see* Gravel	
Phosphate acid, ammoniated	52
fertilizer	60
bone, fertilizer	60
pebble, dry	90
wet	100
rock, broken, dry	75
wet	85
Plumbago, crushed	85–90
Pumice stone, ground	40
Pyrites, *see* Limestone, add 50%	
Quartz, *see* Limestone	
Rock, soft, excavated with shovel	100–110
Salt, coarse	56
fine	45
lump, for stock	100
Saltpeter	68
Sand, beach or river, wet	120–130
dry	90–100
foundry, loose	80–90
rammed	100–110
quartz, dry	110
Shale, broken	90–100
crushed	85–90
Slag, blast furnace, crushed	80–90
granulated, dry	60–65
wet	90–100
Slate, *see* Limestone	
Soap	50
Soda ash, carbonate, dense	50–65
light	30–40
nitrate	85
Sulphur, *see* Limestone, less 25%	

Table C

Miscellaneous Materials Handled by Belt Conveyors and Belt Elevators

	Weight per cubic foot
Bagasse, fresh, moist	7.5
dry, loose	5.0
20 to 25% of weight of sugar cane from which it comes	
Bakelite and other plastics, powdered	30–40
Baking powder	50
Beet pulp, dry	12–15
wet	25–45
Bones, crushed	35–40
Bone black	25
Bonemeal	55–60
Brewer's grains, dry	25–30
wet	55–60
Brick, hard	125
soft	100
Carbon black, powder	5
pellets	25
Char, sugar refinery	45
Chips, paper mill, softwood	12–20
yellow pine	20–25
hogged, fuel	22
Cocoa	30–35
Cocoa beans	33
Cocoa nibs	35
Cocoanut shells, broken	30
Coffee, fresh berry	43
dried berry	40
Compost, hothouse	50
Copra (cocoanut meats)	22
Corn grits	42
Cotton-seed cake, crushed	40–45
Cotton-seed hulls	12
Cotton-seed meal	35
Cotton-seed meats	45
Fat	58
Filter press mud, sugar factory	70
Fish, raw	52–56
Fish meal	40
Flour, wheat, rye	40
Foundry refuse, old sand, cores	60–80
Garbage, household	50
Gas-house oxide, sponge	28–35
Glass, broken, *see* Cullet	
Glue, animal, flaked	35
vegetable, powdered	40
Gluten meal	39
Guano, dry	70
Gum arabic	90
Gunpowder	63
Gutta-percha	60
Hay, loose	5
pressed	8
Hops, brewery, moist	35
Ice, crushed	40
India rubber, raw	58
devulcanized	35
Iron borings, machine shop	125
Lead, white, pigment	250–260
Linseed cake, crushed	48–50
Linseed meal	44
Malt meal	36–40
Malt sprouts	15
Manure, stable	25
Milk, dry, powder	36
malted	27
Mushrooms	24
Oil cake, *see* Linseed cake	
Oxide, gas-house sponge	28–35
Oyster shells	80
Paper pulp	60
Paraffine cake, broken	45
Petroleum coke	35–40
Pitch	72
Potash salts, sylvinit, etc	80
Resins, powdered	30–40
Rice grits	40
Rosin, lump	65
Rubber	58
reclaimed	35
Sawdust	13
Sewage (sludge)	40–50
Skinnings, slaughter-house refuse	75
Slurry, cement	90
Snow, fresh fallen	5–12
compacted by rain	15–50
Soap chips	10

TABLE C—*Continued*

	Weight per cubic foot
Starch	45
Steel turnings	60–120
Sugar, cane stalks	25
raw	55–65
refined	55
bag, Cuban, 27″ × 14″ × 42″	325
Hawaii, 15″ × 30″	100
Philippines 16″ × 12″ × 24″	100–125
Talc, *see* Limestone	
Tallow	58
Tan bark, ground	55
Tankage	40
Trap rock, *see* Limestone, add 15%	
Turf, dry	30
Whiting, *see* Chalk	
Wood chips, *see* Chips	
Zinc ores, crushed	150–160
Zinc oxide	10–30

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